Report 76-13009

LONG LIFE COOLANT PUMP TECHNOLOGY

AiResearch Manufacturing Company of California
A Division of The Garrett Corporation
2525 W. 190th Street
Torrance, California 90509

September 1976

Final Report

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Prepared under Contract NAS8-28434 for
National Aeronautics and Space Administration
George C. Marshall Space Flight Center
Marshall Space Flight Center, Alabama 35612

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FOREWORD

This final report documents the Long Life Coolant Pump Technology Program which was conducted for the NASA George C. Marshall Space Flight Center, Marshall Space Flight Center, Alabama, under Contract NAS-8-28434. The work was performed by the AiResearch Manufacturing Company of California, a Division of the Garrett Corporation, Torrance, California. Mr. Michael D. Leberman was the NASA-MSFC Program Technical Monitor.

SUMMARY

A program to investigate and improve Long Life Coolant Pump Technology was started in August 1972 with the objective of establishing pump technology suitable for space mission durations of two years and greater. This work was performed by the AiResearch Manufacturing Company of California, a Division of the Garrett Corporation, for the NASA George C. Marshall Space Flight Center, under Contract NAS-8-28434. The work was an outgrowth of the Apollo Telescope Mount (ATM) Long Life Coolant Pump Program, and design data from that program were used as a starting point for the work reported here.

Based upon previous work with the ATM pump, design concepts were investigated for improving coolant pump technology, with particular reference to extending the period of reliable life. A key item in these design concepts was an improved bearing system for the pump rotating elements. Designs were prepared and an ATM coolant pump was modified to incorporate such an improved bearing system. This bearing system consisted of double conical bearings, pressurized with the working fluid from pump discharge. After preliminary calibration tests, this pump was satisfactorily endurance tested pumping Freon 21 for 12,304 endurance hours, with testing terminated at the end of the program in October 1976.

In addition, a new, prototype pump was designed, using various improved design concepts, and fabricated as a non-flight weight test article. This prototype pump was satisfactorily endurance tested pumping Freon 21 for 10,382 endurance hours, with testing terminated at the end of the program. In these tests, Freon 21 was selected as the working fluid because it represented a viable candidate as a space system heat transport fluid, and in addition it represented one of the more difficult fluids from the viewpoint of achieving pump life. For instance, pump bearings which operated satisfactorily on Freon 21 could rather easily be converted to operate on the Coolanol's or other typical space system heat transport fluids.

This final report documents the approach, difficulties, and achievements of the Long Life Coolant Pump Technology Program and, in addition, presents recommendations and suggestions regarding possible future efforts.

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(Continued)

1. INTRODUCTION

The AiResearch Manufacturing Company of California, a Division of the Garrett Corporation, conducted an investigation of Long Life Coolant Pump Technology, under NASA-Marshall Space Flight Center Contract No. NAS8-28434. The work was an outgrowth of the Apollo Telescope Mount Coolant Pump Program, and design data from that program were used as a starting point for the work reported here. In the program, concepts were investigated for improving coolant pump technology, with particular reference to extending the period of reliable life. A key item was an improved bearing system for the pump rotating elements. Such an improved bearing system was designed and demonstrated in two different pumps which accumulated 12,304 endurance hours and 10,382 endurance hours respectively, prior to termination of the program.

1.1 Introduction of Problem

Operational experience with the Apollo ECS Coolant Pump and the Apollo Telescope Mount Coolant Pump well demonstrated the soundness of the basic design approach which was common to both pumps. The basic concepts included in this design approach were

- Redundant pumps to optimize system reliability
- Centrifugal type pump design
- Magnetic coupling drive
- Dry motor
- Pump bearings lubricated by the working fluid
- Motor electrical rotor carried on grease packed ball bearings

Figure 1 shows a view of the Apollo Block II ECS Coolant Pump Package, 825070, and Figure 2 shows a view of the Apollo Telescope Mount Coolant Pump Package, 580745. Figure 3 shows an exploded view of the Apollo Telescope Mount Coolant Pump Package, showing the typical design arrangement, including the magnetic coupling drive.

The Apollo pumps were designed for relatively short missions, with the longest design operational requirement being 278 days (6672 hours) for the Apollo Telescope Mount Pump. However, it was expected that future space missions would require coolant pumps which could operate continuously for durations up to three years with high reliability, and this prospect raised two questions:

- What design changes would be necessary to the Apollo type coolant pumps to provide a three year mission capability?
- Were there other pump design concepts which would be more nearly optimum, or which may be required to meet potential differences in future problem statements?

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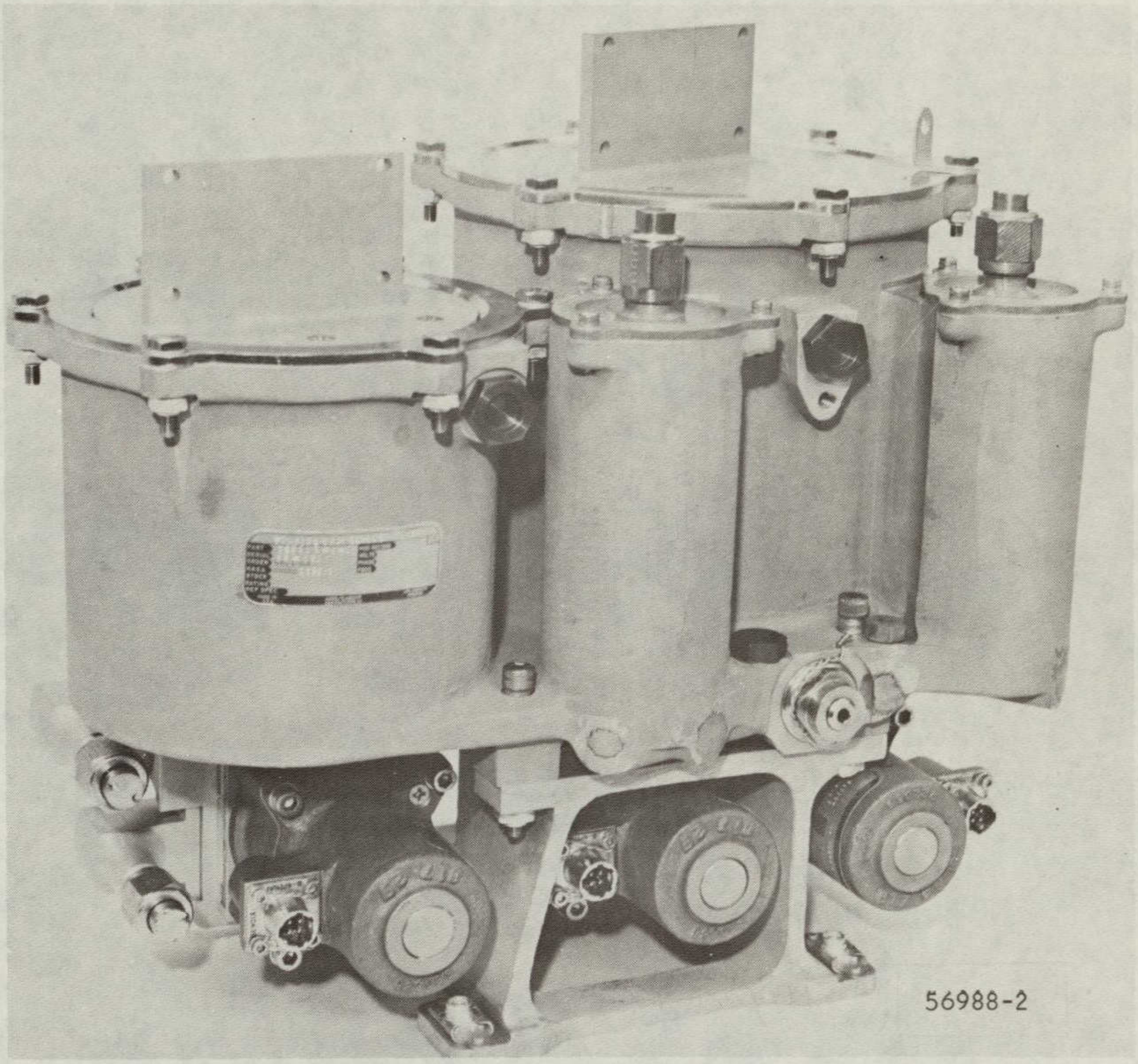

Figure 1-1. Apollo Block II ECS Coolant Pump Package, 825070. Water - Ethylene Glycol Working Fluid, 200 PPH Flow, 36 PSID Pressure Rise. Electrical Input 3 Phase, 115/200 VAC, 400 Hz, 52 Watts Per Pump

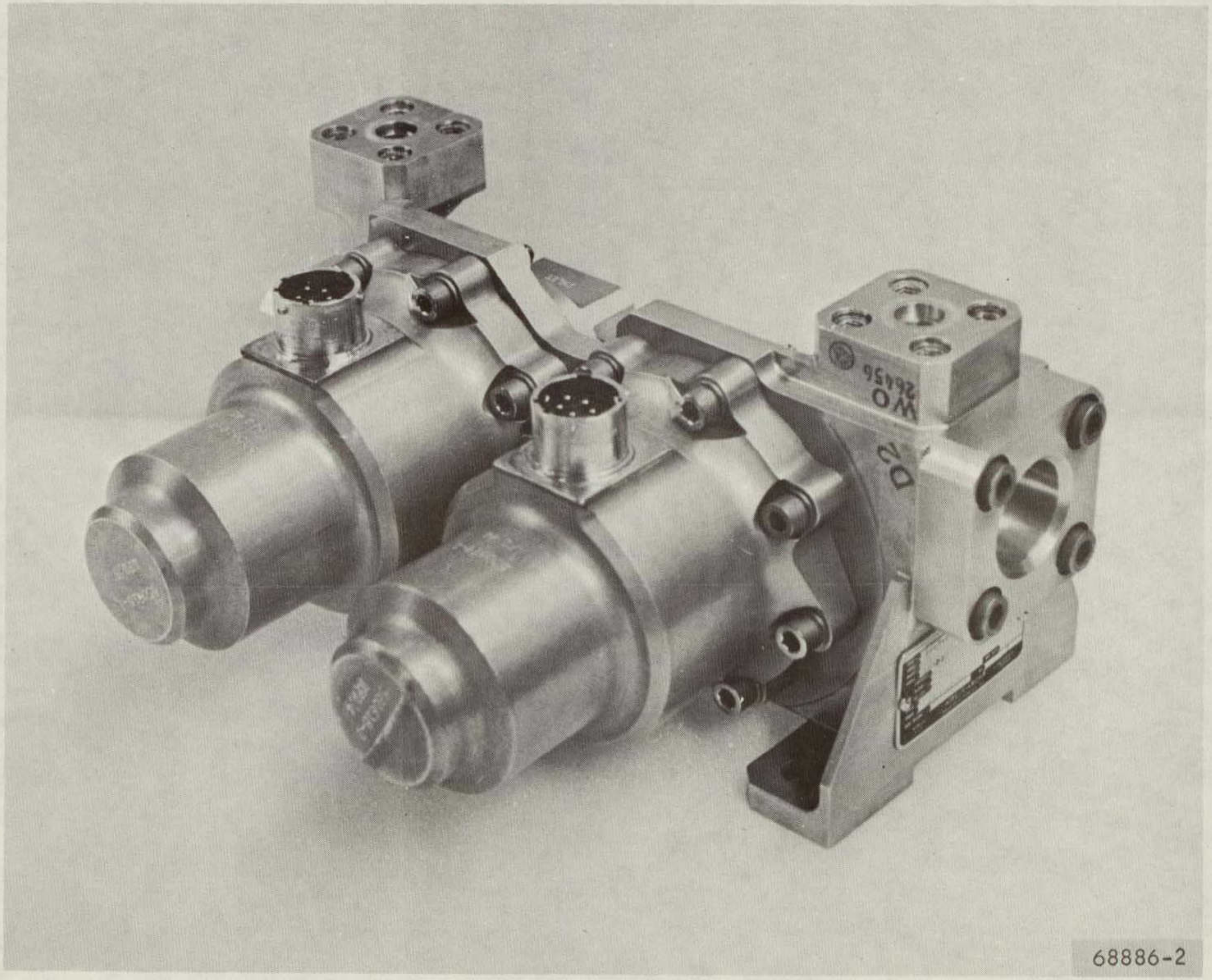

Figure 1-2. Apollo Telescope Mount Coolant Pump Package, 580745. Water Methanol Working Fluid, 900 PPH Flow, 31 PSID Pressure Rise. Electrical Input 3 Phase, 12.2/21.2 VAC, 400 Hz, Quasi-Sine Wave, 125 Watts Per Pump.

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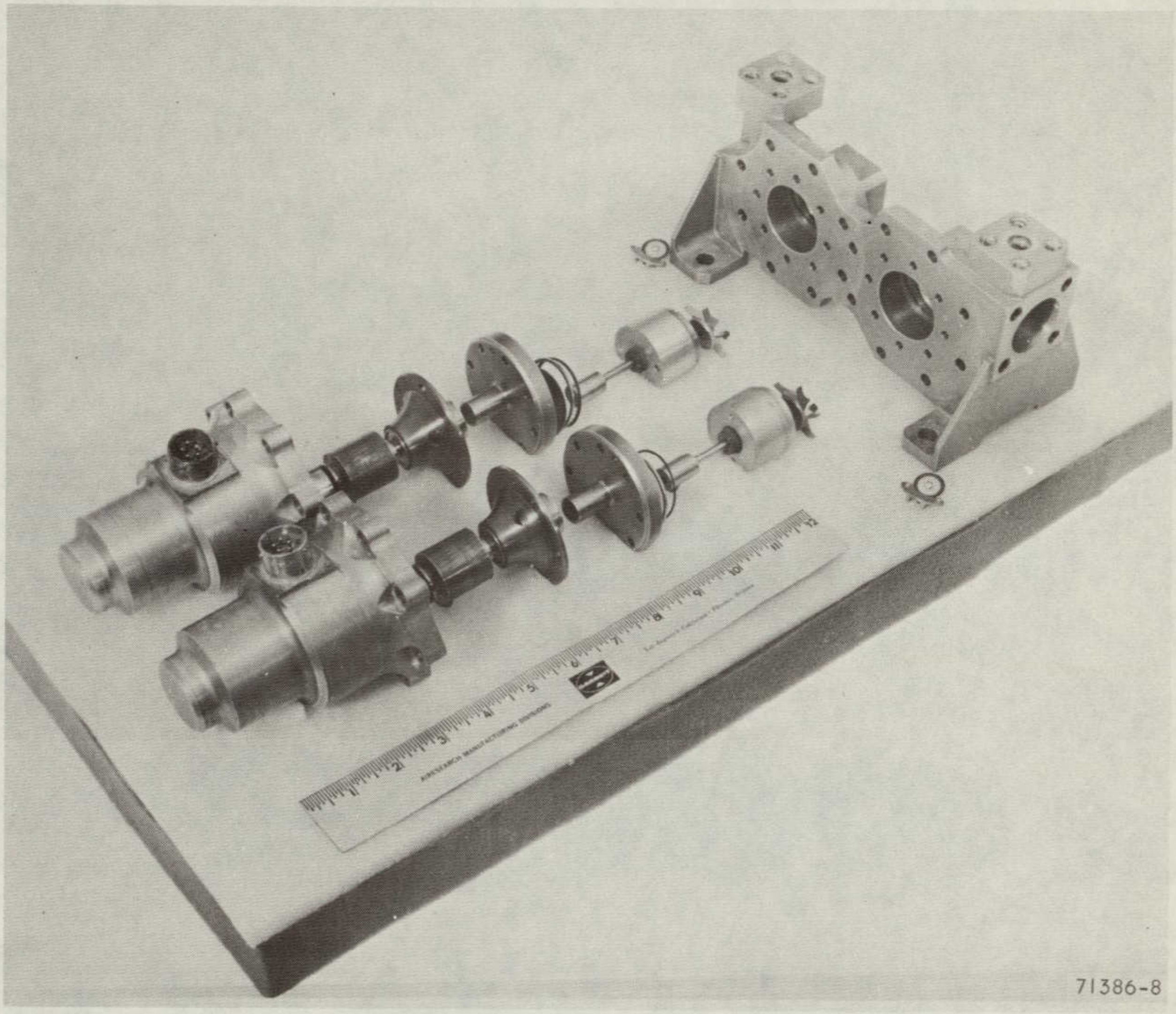

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Figure 1-3. Exploded View of Apollo Telescope Mount Cooling Pump Package, 580745. Showing Typical Design Arrangement, and Magnetic Coupling Drive.

These questions led to the initiation of the Long Life Coolant Pump Technology Program, which had the following objective.

1.2 Program Objective

The overall objective of the program was to perform a theoretical and experimental investigation to improve electrically driven long life coolant pumps for space and cabin environments. Design areas of particular concern were:

- The potential for extended operational life at high reliability
- Optimization of pump performance
- Design flexibility, to accommodate various potential applications
- Serviceability

Primary emphasis in the program was to be invested in achieving the pump life and reliability objective.

1.3 Program Approach

The program approach included three major tasks:

- A conceptual design study starting with definition of the boundaries of the investigation, and including a search for and identification of promising design concepts having the potential for meeting the program objectives. This phase culminated in a conceptual design review with NASA-MSFC, to mutually select promising concepts for further study.
- Detailed analysis of the selected concepts to permit selection, with NASA-MSFC, of an optimum concept to be implemented in hardware. Preparation of detailed drawings and fabrication of a demonstrator pump unit.
- Development testing of the demonstrator pump in areas of concern, plus endurance testing, within the limits of the available funding.

The program drew heavily on AiResearch past experience in the design, development, and production of advanced technology pumps, motors, and controls for spacecraft applications. In addition, the program was a logical continuation of the NASA-MSFC program for design, development and qualification of the Apollo Telescope Mount Coolant Pump, and design data from that program were used as a starting point for the work reported here.

1.4 Program Accomplishments

Accomplishments of the program were as follows:

- A hydrodynamic problem statement was established for a future "typical" space system coolant pump

Freon 21 heat transport fluid.
100°F pump inlet temperature.
Pump inlet pressure as required.
2400 PPH pump flow.
60 PSID pump pressure rise.

- Alternative hydrodynamic configurations of different manufacturing complexity were established and the hydrodynamic performance was estimated. A simple configuration was selected to fabricate for test demonstration in a prototype pump, although more complex configurations offered the potential for increased pump efficiency.

- Alternative bearing configurations were considered for the pump rotating group. The pump rotating group bearings were the highest technical risk item in the previous pump experience. A pressurized double conical bearing configuration was selected for development, and demonstration in a prototype pump.

- An ATM pump was modified to include the pressurized double conical bearing for the pump rotating group, to provide early demonstration of the feasibility of this concept. This modified ATM pump was endurance tested for 12,304 hours, with Freon 21 working fluid. No deterioration in pump performance or significant operating parameters was noted during the endurance test.

- A somewhat larger prototype pump was designed and fabricated, using the pressurized double conical bearing for the pump rotating group, and endurance tested for 10,382 hours with Freon 21 working fluid. No deterioration in pump performance or significant operating parameters was noted during the endurance test.

1.5 Conclusions

Conclusions made at the end of the program were:

- The all metal pressurized double conical bearing was shown to be feasible for long life coolant pumps, and a decided advantage for use with fluids having a tendency to swell or otherwise dimensionally distort non-metallic materials.

- Some improvement in pump hydrodynamic efficiency is achievable with configurations having greater manufacturing complexity. Such configurations may be warranted for power limited space applications.

- The magnetic coupling approach continues to demonstrate its viability as a means of eliminating shaft dynamic seals, of isolating the motor elements from fluids which may be active solvents such as Freon 21, and of minimizing rotating element fluid viscous losses, compared to "canned" motor approaches.

- Grease packed bearings for the motor electrical rotor, with a properly designed installation, show very little change after 12,000 hours operation.

- Based upon an overall assessment of the results of the program, it is concluded that reliable coolant pumps can be designed for three year space missions (26,280 operating hours) with a high confidence level.

1.6 Recommendations

Based upon the results of the investigation and upon the current state of several other significant technological developments, the following actions are recommended:

1.6.1 Flight Version of Prototype Pump

Prepare a flight weight design of the prototype pump, fabricate, and test in the significant areas which were not covered in the program to date. These additional test areas include vibration and acceleration tests, plus any other significant tests required for a particular intended application. Such a program, representing a modest amount of additional effort, would provide a coolant pump design of demonstrated life capability, suitable for long duration space missions.

1.6.2 Continued Advanced Technology Program

Several recently developed advanced design concepts have the potential for significantly reducing typical space vehicle coolant pump size, weight, and electrical input power, while retaining a long life capability. Such improvements naturally would have a significant impact on the design of space vehicle coolant systems. Accordingly, it is recommended that these concepts be evaluated and demonstrated in a "next generation" prototype design pump, to establish and verify their range of applicability. These recommendations include:

- Investigate the use of a brushless DC motor of advanced design, possibly operating at a higher rotational speed than that representative of current coolant pump design practice. This recommendation is based upon recent advances in brushless DC motor technology, and in solid state electronics, which have produced highly efficient motor systems having excellent reliability.

  The brushless DC motor would be a permanent magnet type, having a basic motor efficiency of 0.91 compared to the 0.65 obtainable with a 3 phase a-c motor in the size range typical for space system coolant pumps.

  The efficiency of the front end electronics for the brushless DC motor is approximately 0.82, which is equivalent to the efficiency of the DC to AC inverter required for AC motors operating on the typical space vehicle DC power system. The resulting efficiency

product of the electronics plus motor is then 0.75 for the brushless DC system, and 0.53 for the 3 phase AC system. This represents a 42% improvement in the efficiency of the motor and electronics, obtained by using the brushless DC motor.

The brushless DC motor can operate at a preferred rotational speed determined by the pump hydrodynamics, and is not constrained to certain particular speeds determined by the number of motor poles and the electrical input frequency, as is the AC induction motor. In some cases this can be a distinct advantage if the hydrodynamic requirements place the pump optimum specific speed (best efficiency design) and desired rotational speed at a speed not readily achieved by the AC induction motor operating at the available frequency. Also, it should be noted that the motor weight and size are approximately inversely proportional to rotational speed, presenting the potential for significant reductions in size and weight, if it should prove feasible to operate at higher design rotational speeds.

In addition, the speed of the brushless DC motor can be varied easily by changing voltage level. This offers the potential for modulating system cooling as required, by varying pump speed, while saving electrical power during those periods when maximum cooling flow is not desired.

Another advantage is that the brushless DC motor does not have a significant amount of heat generated in the rotor, as AC motors have. This greatly simplifies the problem of keeping the motor bearings cool, which is one of the most significant requirements for long life of the motor grease packed bearings.

- Investigate designing the pump hydrodynamics for a somewhat higher specific speed. A specific speed of approximately 1500 is recommended, and this shows a higher theoretical efficiency. However, it results in a smaller diameter, higher speed, impeller and the "normal" estimate of the reduction in efficiency due to the size scale effect predicts that the higher efficiency will not be achieved with normal manufacturing techniques. The use of improved manufacturing techniques and much closer control of the geometric tolerances of the pump hydrodynamic elements offers the potential of significantly reducing the magnitude of the size scale effect, permitting the actual achievement of a higher efficiency coolant pump of smaller size. The higher manufacturing cost of such a pump may be warranted by power limited space vehicle applications.

- Investigate the use of an advanced samarium cobalt alloy for the magnetic coupling, instead of platinum cobalt. The samarium cobalt alloy has a magnetic strength maximum energy product (BH_{max}) of 23×10^6 Gauss-Oersteds, compared to 9.5×10^6 Gauss-Oersteds for the platinum cobalt alloy. This permits reducing the size parameter (D^2L) of the magnet system in direct proportion to the increase in energy product, i.e., a reduction of 59%. Such a reduction in magnet dimensions significantly reduces the viscous loss associated with the pump magnet rotating in the working fluid, and it is expected that a worthwhile increase in pump overall efficiency can be achieved.

The samarium cobalt alloy is more active chemically than the platinum cobalt, so a protective coating or plating will be required. A cost advantage is the fact that the samarium cobalt alloy is less expensive than the platinum cobalt by a factor of 15, and this can result in a very significant reduction in pump cost.

- Investigate further design improvements to the pressurized bearing used for the pump rotating group. In particular, the pressurized journal bearing and flat configuration thrust bearing should be evaluated in comparison with the double conical configuration, since it may significantly reduce pump break-away torque, thus improving the starting operational margin. In addition, it has the potential for reducing the sensitivity of pump bearing operation to pump bearing system axial clearance, and it would simplify manufacture.

Properly worked out and integrated into a pump design, these advanced design concepts offer the potential for achieving a superior "next generation" coolant pump for long duration space missions.

2. DESIGN CONCEPTS STUDY

2.1 Objectives

The objective of the Long Life Coolant Pump Technology Program was to establish pump technology suitable for space mission durations of two years and greater, starting from the existing technology base which had been established in the previous programs, particularly the Apollo Telescope Mount (ATM) Coolant Pump Program.

The overriding design criterion in the investigation was to achieve technology improvements which would extend the period of reliable operating life, since this was the only missing element for the application of existing pumps to multi-year missions. Secondary design criteria also considered in the study included:

- Improved efficiency through performance optimization.
- Design flexibility regarding flow and pressure rise.
- Pump compatibility with various working fluids.
- Serviceability.

In the ATM pump program, the non-metallic pump element bearings presented the biggest development problems, particularly in regard to the dimensional changes caused by immersion in the methanol-water working fluid. Accordingly, a specific study objective was to achieve a pump bearing system which would not be subject to dimensional changes resulting from such exposure to the working fluid, regardless of the fluid which was ultimately selected.

2.2 Approach

The approach used in the design concepts study was to postulate a problem statement for a future space vehicle coolant pump, and then to consider alternative design approaches for meeting this problem statement, keeping in mind the ranking of design criteria discussed above. A selection was then made of a design approach judged to be a best candidate for future development.

The pump basic problem statement which was postulated for the study included:

- Freon 21 working fluid. This fluid has good heat transport properties, and requires low pumping power because of its low viscosity and high density, and hence is a strong candidate for space vehicle heat transport. In addition, Freon 21 represents one of the more difficult fluids from a pump immersed bearing viewpoint, and a bearing developed to operate satisfactorily on Freon 21 could rather easily be adapted for use with other fluids.

- Pump flow 2400 PPH and pressure rise 60 PSID. These values were selected because they were typical of the requirements of several NASA space vehicle systems being considered at that time.

- Pump inlet temperature 100°F. This was selected as typical of the requirements of several NASA space vehicle systems being considered at the time.

- Pump inlet pressure; as required. That is, design freedom was given to this item, and it would then become a specification item for the remainder of the coolant system design.

- Pump overall efficiency, at least as good as existing designs.

- Pump size and weight, competitive with existing designs.

- Electrical input power, selected as 3 phase, 400 Hz, 115/200 VAC for study convenience. It was recognized that the newer technology in brushless DC motors should also be studied, but it was defined as outside of the scope of the present effort.

- Operating life, two years minimum, at high reliability.

2.3 General Design Arrangements

Coolant pump general design arrangements were reviewed, with resulting considerations as follows:

- In regard to the question of wet motor vs dry motor, the dry motor was selected because it can be designed to yield a significantly higher overall efficiency than the wet motor. The wet motor is subject to the viscous losses of the rotor running in the working fluid. In addition, for the more active fluids as Freon 21 (which is a very good solvent), long life reliability would probably require a metallic bore liner to keep the fluid from the stator insulation. Such a bore liner increases the magnetic "air gap" and reduces motor efficiency.

- The question of magnetic coupling drive vs dynamic shaft seal was resolved in favor of the magnetic coupling drive. For long duration missions of systems containing only a small amount of coolant, even a slight leak presents a possibility of running out of fluid and loss of the mission. The magnetic coupling drive provides a fluid system which is essentially hermetically sealed, and has proved to be reliable in other space system coolant pumps. It eliminates the significant hazard of a leak from a dynamic shaft seal. The magnetic coupling drive does require pump rotating group bearings which operate in the working fluid. However, it was considered that a reliable solution to this problem could be found, while the use of dynamic shaft seal would always pose a significant element of risk.

- The question of pump type was resolved in favor of the centrifugal type hydrodynamic pump, because it provides good efficiency and has no sliding contacts which can cause frictional hang-up during starts, and which can generate wear debris when running. The vane pump and the gear pump can be adapted to the space coolant pump requirement, but have the sliding contact problem.

2.4 Electric Motor Design

2.4.1 Electric Components

The motor design was selected as a 3 phase AC induction motor, with a wound stator and cast integral rotor. This was representative of existing state-of-the-art which could provide the long life capability without further refinement.

2.4.2 Motor Bearing System

The motor bearing system was considered to be a key item in achieving the desired extended operating life at high reliability. One candidate was the grease packed ball bearing, which is a reliable state-of-the-art item and which has the potential for providing long life in properly designed installations. However, the grease does have a limited life, and the life is significantly influenced by operating temperature. If kept suitably cool, the grease life can be extended for long missions. In addition, extra wide bearings ("cartridge width") can be used to provide a 50% increase in initial grease capacity, thus further extending the potential operating life.

Another candidate for motor bearings for long duration missions is the foil type gas film bearing, which does not have a lubricant life problem. However, the gas film bearing has a significantly higher starting torque, and is better suited for operation at higher rotational speeds than that of the typical coolant pump. In addition, use in a hard vacuum space environment would require hermetic sealing of the electric motor to retain sufficient gas density to provide an adequate bearing film, and the bearing operation would be extremely vulnerable to leaks in this hermetic seal. For these reasons, cartridge width grease packed bearings were selected for the motor rotor bearings.

2.4.3 Motor Thermal Design

Two objectives were paramount in the thermal design of the motor: keep the motor bearings cool and keep the stator winding cool. These requirements were established because of the close relationship between bearing grease life and bearing operating temperature, and between stator winding insulation life and stator winding temperature. These objectives were achieved by designing to utilize the pump working fluid as a sink for the motor heat, and the detail thermal design was carried out by computerized analysis. Appendix A shows the thermal analysis for the prototype pump 581280, and shows coolant passages which were initially provided around the stator housing, but which were later determined to be unnecessary. A significant factor in the thermal design to keep the motor bearings cool

was the use of bearing supports made of high conductivity beryllium copper alloy.

2.5 Pump Design

2.5.1 Pump Hydrodynamic Design and Performance

The pump hydrodynamic design was investigated for three different design speeds and three different type geometries, as shown in the following matrix:

Configuration	Rotational Speed, RPM		
Shrouded Impeller, 3 Dimensional Blades	7,766	11,600	23,300
Shrouded Impeller, 2 Dimensional Blades	7,766	11,600	23,300
Unshrouded Impeller, 2 Dimensional Blades	7,766	11,600	23,300

Table 2-1 shows the nomenclature used in the analysis and Figure 2-1 shows the pressure-enthalpy diagram for the Freon 21 working fluid.

Table 2-2 shows the pump performance parameters for the shrouded impeller configuration with 3 dimensional blades, and Figure 2-2 shows a plot of the estimated performance.

Table 2-3 shows the pump performance parameters for the shrouded impeller configuration with 2 dimensional blades, and Figure 2-3 shows a plot of the estimated performance.

Table 2-4 shows the pump performance parameters for the unshrouded impeller configuration with 2 dimensional blades, and Figure 2-4 shows a plot of the estimated performance.

Figure 2-5 shows a schematic diagram of the shrouded impeller configuration, compared to the unshrouded impeller configuration.

Review of the estimated performance data shows that the best efficiency is obtained with a shrouded impeller design having 3 dimensional blades, and operating at 11,600 rpm. However, for the purposes of the immediate program having a major emphasis on pump bearing development, the simpler unshrouded impeller design with 2 dimensional blades, and operating at the same 11,600 rpm was selected for fabrication in a prototype pump.

Table 2-I

NOMENCLATURE USED IN HYDRODYNAMIC ANALYSIS

N	=	Rotational Speed, rpm
Q	=	Flow, gpm
H	=	Head, ft
NPSH	=	Net Positive Suction Head, ft.
NPSP	=	Net Positive Suction Pressure, psi
D_2	=	Impeller Diameter, in.
U_2	=	Impeller Tip Speed, fps
g	=	Acceleration of Gravity = 32.174 ft/sec^2
ψ	=	$\frac{gH}{U_2^2}$ = Head Coefficient, Dimensionless
N_s	=	$\frac{NQ^{1/2}}{H^{3/4}}$ = Specific Speed
S	=	$\frac{NQ^{1/2}}{(NPSH)^{3/4}}$ = Suction Specific Speed
ν	=	Kinematic Viscosity, ft^2/sec
Re_2	=	$\frac{DU_2}{12\nu}$ = Reynold's Number, Dimensionless
η_o	=	Hydrodynamic Efficiency, Uncorrected
η^*	=	Hydrodynamic Efficiency, Corrected for Re_2
η	=	Hydrodynamic Efficiency, Corrected for both Re_2 and Size

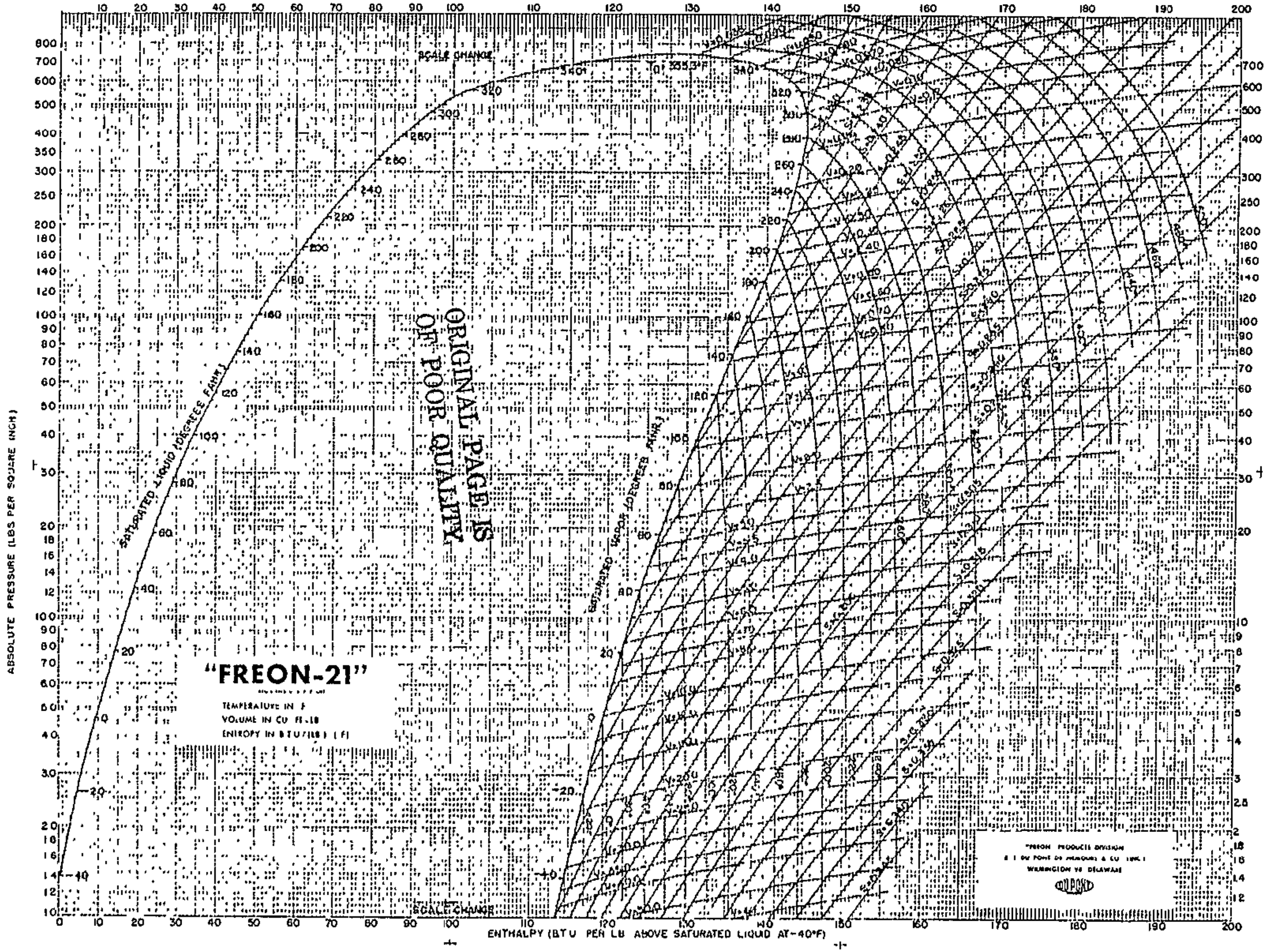


Figure 2-1. Pressure-Enthalpy Diagram for Freon 21.

TABLE 2-2

PUMP PERFORMANCE PARAMETERS,
SHROUDED-IMPELLER, THREE-DIMENSIONAL BLADES

N, rpm	23,300	11,600	7766
N_s	1359	679	453
ψ	0.53	0.59	0.60
U_2, fps	79.4	75.5	74.6
D_2, in.	0.78	1.49	2.20
Re_2	1.64×10^6	2.99×10^6	4.36×10^6
η_0	0.893	0.806	0.716
$1-\eta_0$	0.107	0.194	0.284
$1-\eta^*$	0.128	0.219	0.309
η^*	0.872	0.781	0.691
η/η^*	0.753	0.885	0.939
η	0.656	0.691	0.649
SHP_I	0.192	0.182	0.194
S	9600	9600	9600
NPSH, ft	7.69	3.02	1.77
NPSP, psi	4.45	1.75	1.02

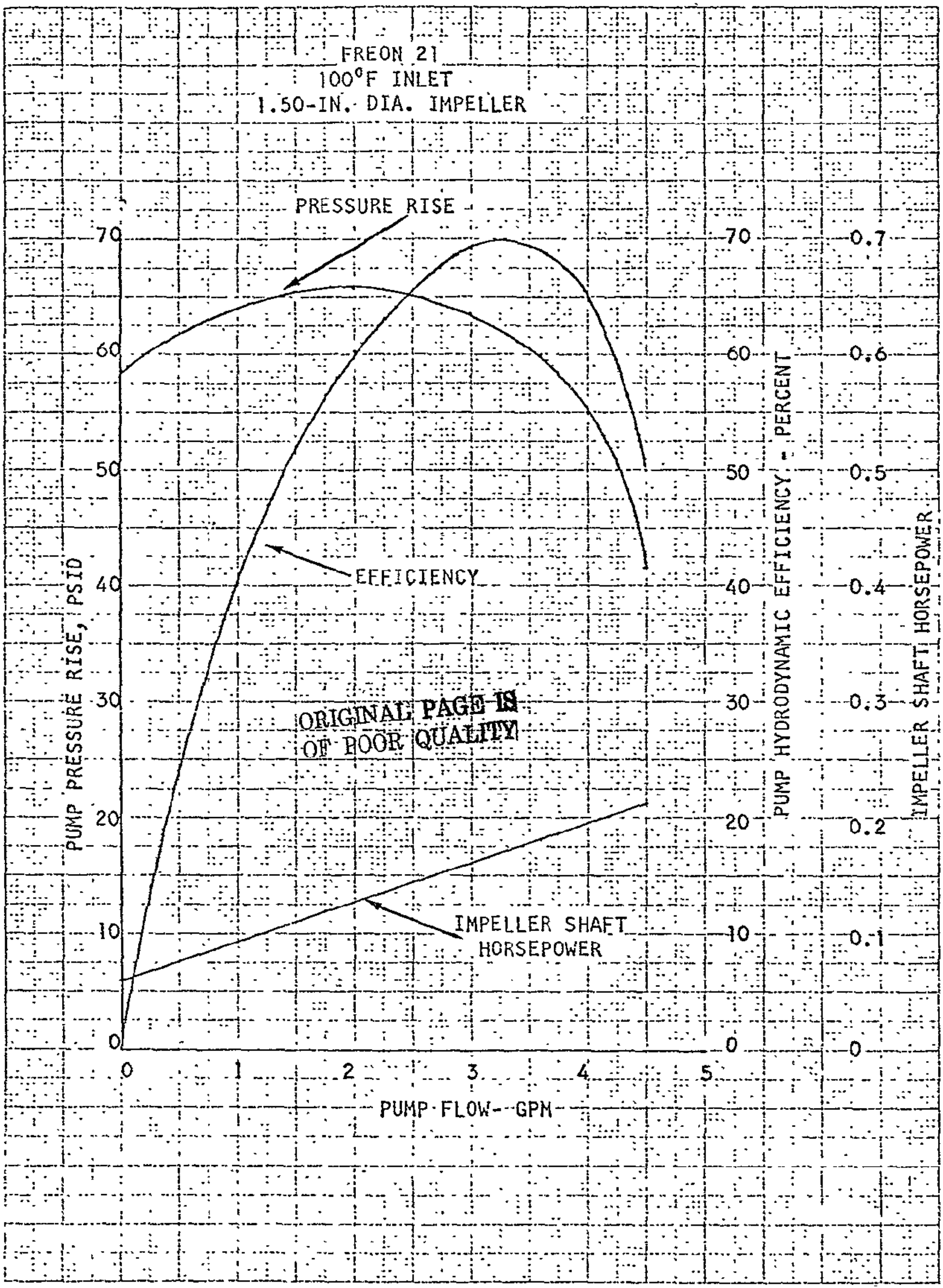


Figure 2-2. Pump Performance with Shrouded Impeller, Three-Dimensional Blades.

TABLE 2-3

PUMP PERFORMANCE PARAMETERS
SHROUDED-IMPELLER, TWO-DIMENSIONAL BLADES

N, rpm	23,300	11,600	7766
N_s	1359	679	453
ψ	0.53	0.59	0.60
U_2, fps	79.4	75.5	74.6
D_2, in.	0.78	1.49	2.20
Re_2	1.64×10^6	2.99×10^6	4.36×10^6
η_0	0.822	0.721	0.594
$1-\eta_0$	0.178	0.279	0.406
$1-\eta^*$	0.213	0.315	0.441
η^*	0.787	0.685	0.559
η/η^*	0.753	0.885	0.939
η	0.592	0.606	0.525
SHP_I	0.213	0.208	0.240
S	8000	8000	8000
NPSH, ft	9.65	3.85	2.25
NPSP, psi	5.58	2.22	1.31

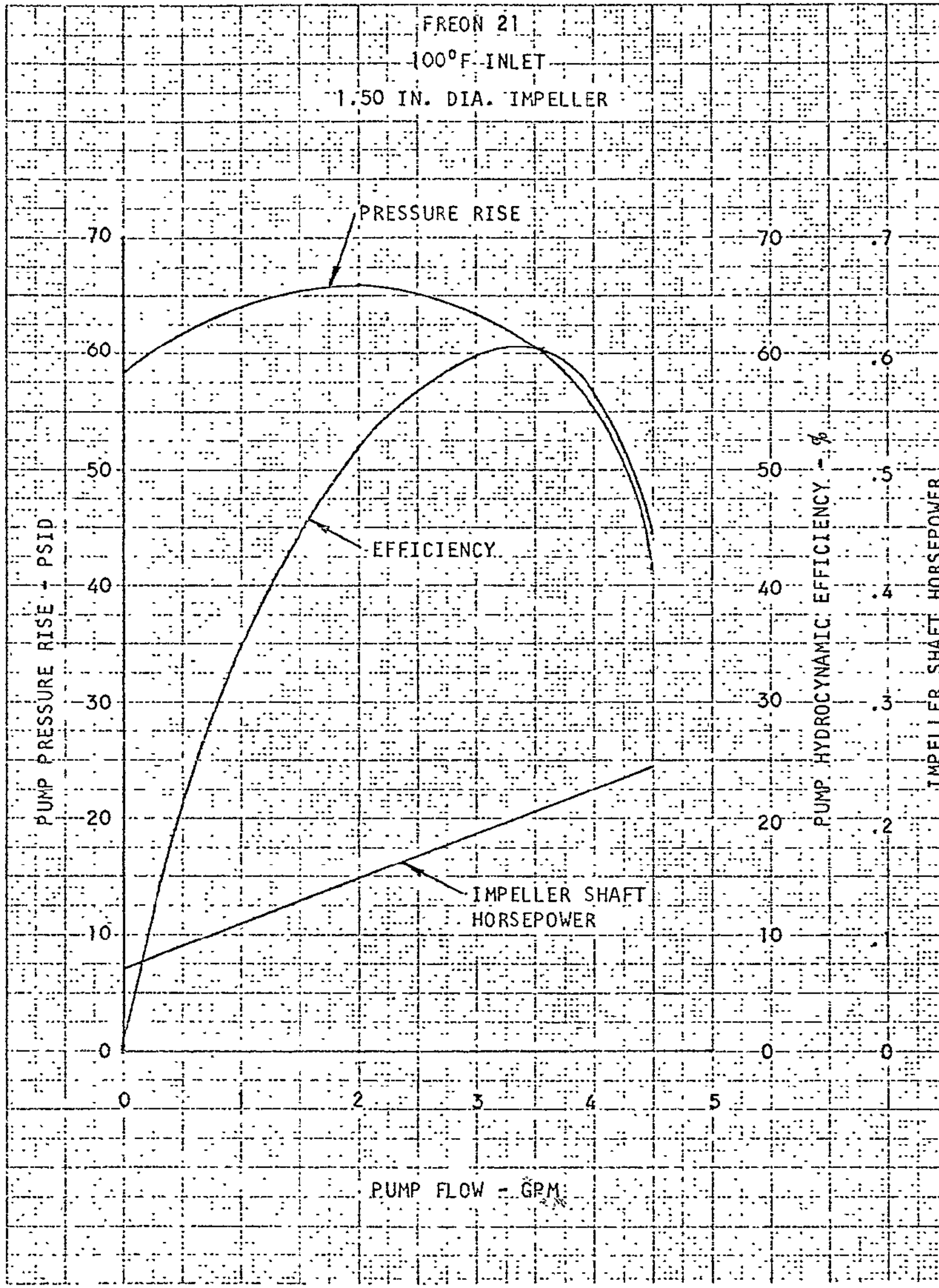


Figure 2-3. Pump Performance with Shrouded-Impeller, Two-Dimensional Blades.

TABLE 2-4

PUMP PERFORMANCE PARAMETERS
UNSHROUDED-IMPELLER, TWO-DIMENSIONAL BLADES

N, rpm	23300	11,600	7766
N_s	1359	679	453
ψ	0.48	0.54	0.55
U_2, fps	83.8	79.0	78.1
D_2, in.	0.82	1.57	2.30
Re_2	1.81×10^6	3.32×10^6	4.76×10^6
η_0	0.742	0.641	0.514
$1-\eta_0$	0.258	0.359	0.486
$1-\eta^*$	0.306	0.401	0.523
η^*	0.694	0.599	0.477
η/η^*	0.760	0.890	0.955
η	0.527	0.533	0.456
SHP_I	0.239	0.236	0.276
S	8000	8000	8000
NPSH, ft	9.65	3.85	2.25
NPSP, psi	5.58	2.22	1.31

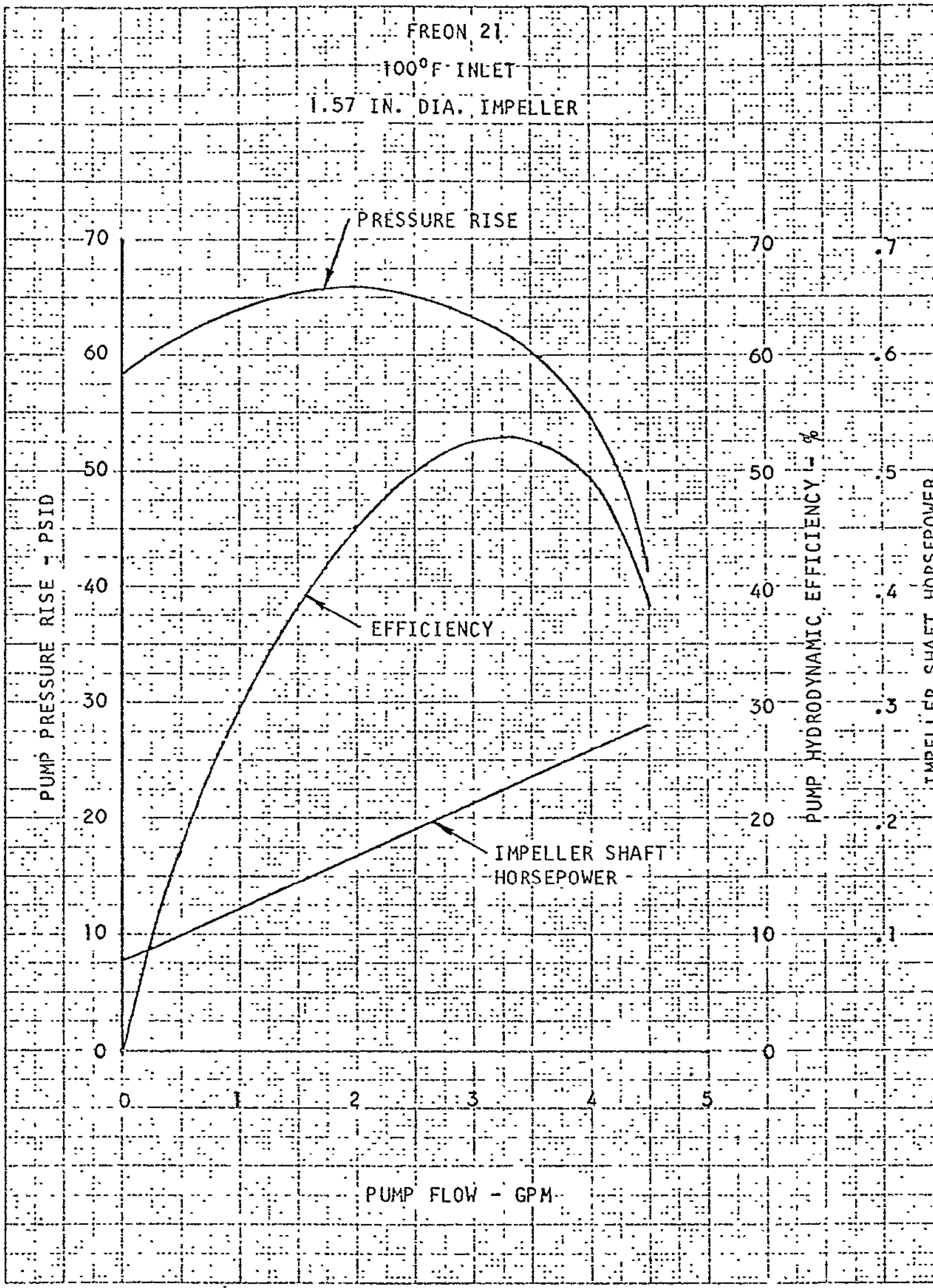
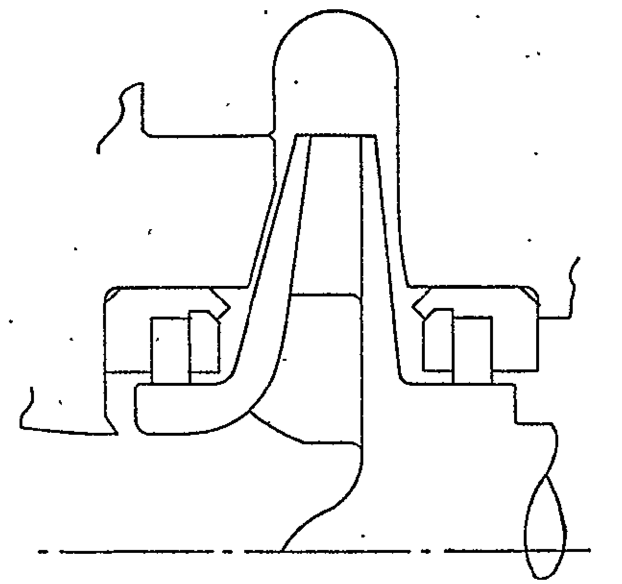
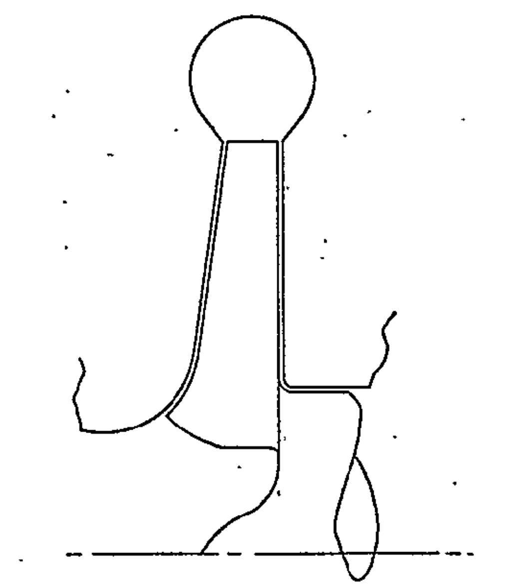


Figure 2-4. Pump Performance with Unshrouded-Impeller, Two-Dimensional Blades.



A: SHROUDED IMPELLER
2 DIMENSIONAL BLADES
6 FULL BLADES
6 SPLITTER BLADES



B. UNSHROUDED IMPELLER
2 DIMENSIONAL BLADES
6 FULL BLADES

Figure 2-5. Showing Alternative Hydrodynamic Design Configurations.

2.5.2 Pump Bearing System

The bearing system for the pump rotating group was considered to be the most critical item for achieving the desired capability of extended life at high reliability. Because of previous experience with dimensional changes of non-metallic bearings, caused by immersion in the working fluid, it was judged that the bearing materials must be essentially metallic or ceramic, with at most a thin non-metallic coating for boundary lubrication during starts and stops. Rolling element bearings were ruled out as having too high a power loss when operated fully immersed. It was considered that pressurizing of the bearing with working fluid from the pump discharge would be desirable to help insure a fluid film during all operating conditions other than starts and stops. Pressurized bearings of straight journal configuration were considered as shown in SK 65619, Sheet 2. These were considered in both the hydrostatic and non-hydrostatic configurations. A final choice was made of the pressurized, hydrostatic, double conical bearing, shown schematically in SK 65619, Sheet 1, and in Figure 2-6. The hydrostatic, double conical bearing was shown in the literature to provide stable operation in the selected configuration. In addition, it has both a radial and axial load carrying ability. Fluid to pressurize the bearing was taken from the impeller cavity at a radius approximately equal to the impeller outer radius, fed to hydrostatic pockets in the female cone and returned to the impeller cavity at a smaller radius. Both the female cone and male cone were to be made of metals, selected for compatible expansion coefficients, and a thin teflon coating was to be applied to the female cone to provide boundary lubrication during starts and stops. It was this configuration which later was successfully demonstrated in a modified ATM pump, 580745, and a prototype pump, 581280.

2.6 Magnetic Coupling Design

The magnetic coupling drive, which eliminates the requirement for a dynamic shaft seal, is shown schematically in SK 65619, Sheet 1. A continuous ring permanent magnet, having numerous radial poles, is mounted on the motor drive shaft, and is termed the drive magnet. A similar continuous ring permanent magnet with radial poles is mounted on the pump shaft, and is termed the driven magnet. In the annular space between the two magnets a thin walled, metallic fluid barrier is used to retain the fluid in the pump cavity. The coupling is a no-slip design and is sized to provide a break-away torque in excess of the motor pull-out torque, thus precluding the loss of drive torque during any mode of pump operation. Platinum cobalt alloy was selected for the permanent magnets, and stainless steel was used for the thin walled fluid barrier. This basic design approach has been used successfully in a variety of space vehicle coolant pumps.

DWG NO SK 65619 | REV LTR

REVISIONS			
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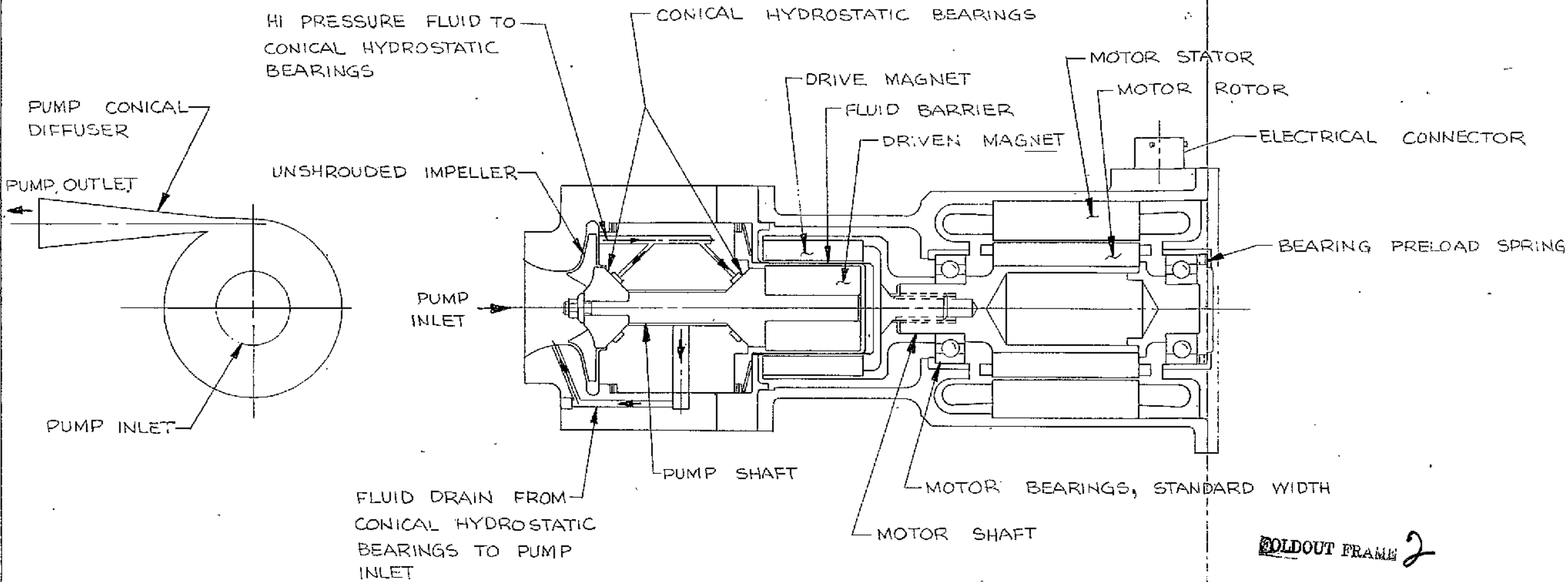


FOLDOUT FRAME 2

FOLDOUT FRAME 1

QTY REQD	ITEM NO.	CODE IDENT NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	SYM
	← ASSY			PARTS LIST	

REQD	NEXT ASSY	USED ON
APPLICATION		

UNLESS OTHERWISE SPECIFIED: STD INTERPRETATIONS PER PIB5 BURR CONTROL PER SC653

IDENTIFICATION MARKING PER MC16

PROCESS

HEAT TREATMENT

CONTRACT NO.

DFT Koon... 6-11-73

CHK

VALUE ENGR

APPD R... 6-11-73

AIRESEARCH APPD

OTHER ACTIVITY APPD

AIRESEARCH MANUFACTURING COMPANY
A DIVISION OF THE GARRETT CORPORATION
LOS ANGELES, CALIFORNIA

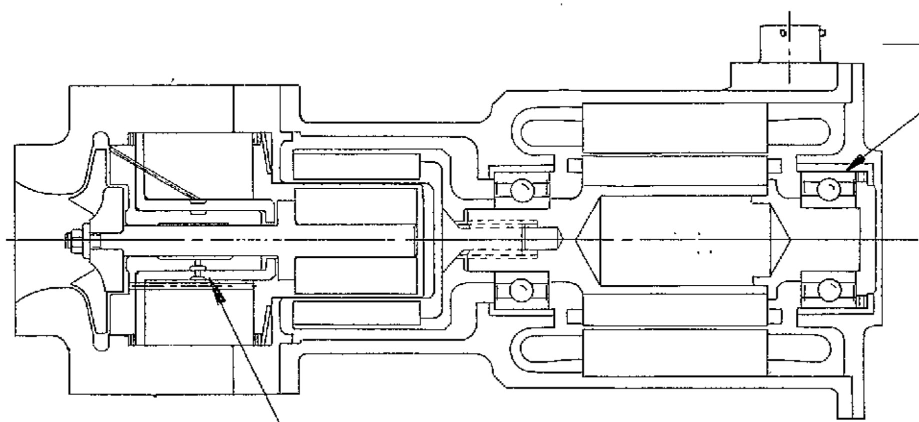
FREON 21 PUMP, MOTOR DRIVEN

SIZE B | CODE IDENT NO. 70210 | DWG NO. SK 65619

SCALE 1/1 | SHEET 1 OF 2

REVISIONS

LTR	DESCRIPTION	DATE	APPROVED



ALTERNATE BEARINGS FOR PUMP AND MOTOR

QTY REQD	ITEM NO.	CODE IDENT NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	SYM
	← ASSY			PARTS LIST	

REQD	NEXT ASSY	USED ON
APPLICATION		

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AIRESEARCH MANUFACTURING COMPANY
A DIVISION OF THE GARRETT CORPORATION
LOS ANGELES, CALIFORNIA

FREON 21 PUMP, MOTOR DRIVEN

SIZE B | CODE IDENT NO. 70210 | DWG NO. SK 65619
SCALE 1/1 | SHEET 2 OF 2

FOLDOUT FRAME 1

FOLDOUT FRAME 2

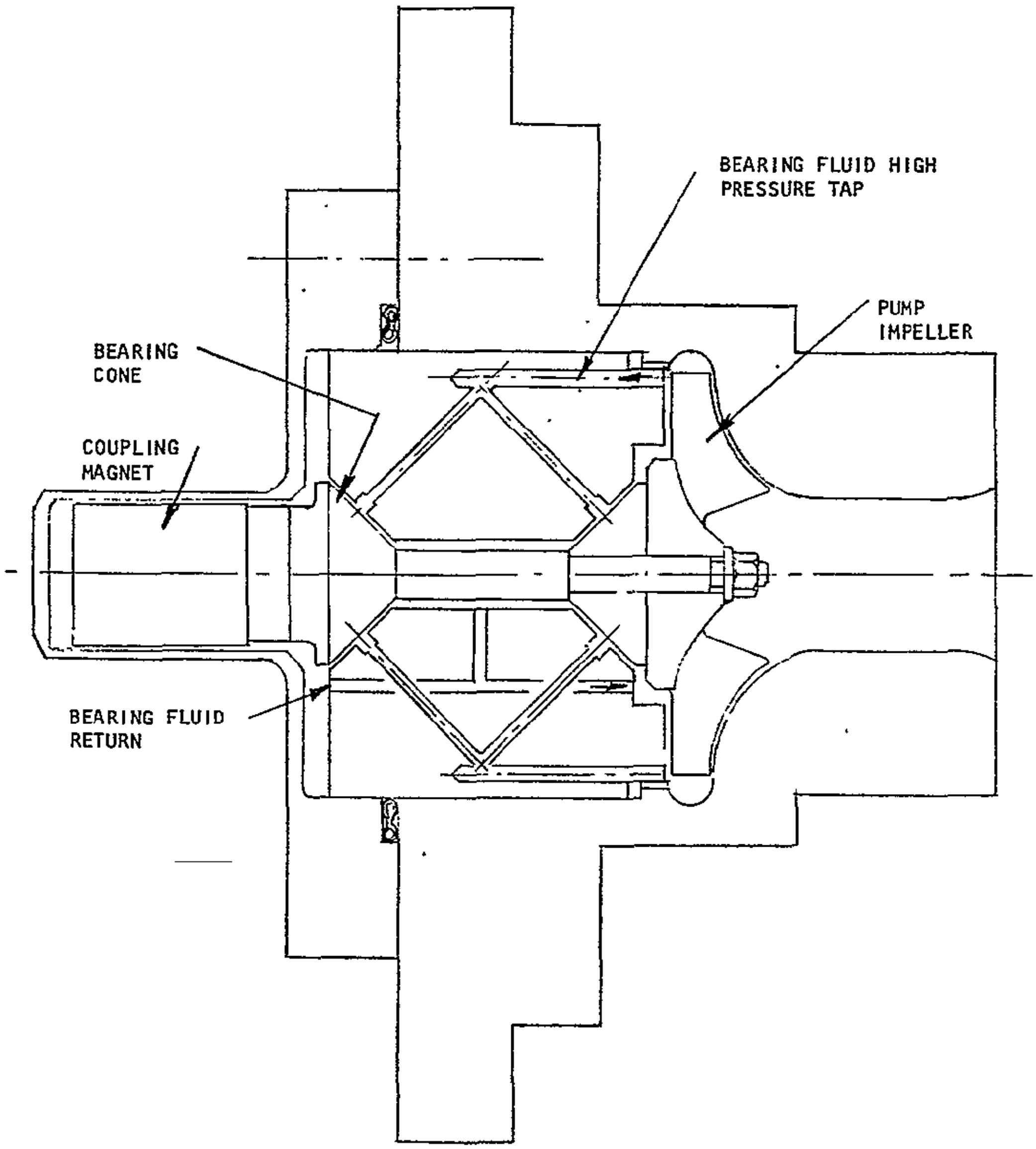


Figure 2-6. Schematic Diagram of Pressurized Double Conical Bearing for Pump Rotating Group.

2.7 Other Items of Detail Design

2.7.1 Materials

Materials selected as suitable for the space system coolant pump application were:

Pump housing	-	6061 T6 aluminum alloy, anodized
Impeller	-	303 S CRES
Bearing cone	-	Inconel 706, Inconel 718; chrome plated
Pump shaft	-	17-4 PH CRES; cone chrome plated
Bearing housing	-	Inconel 706, Inconel 718
Magnets	-	Platinum cobalt alloy
Fluid barrier	-	Inconel 718
Motor housing	-	6061 T6 aluminum alloy, anodized
Motor shaft	-	17-4 PH
Electrical rotor	-	17-4 PH CRES, OFHC Copper, Electrical steel AISI-M19C-4
Electrical stator	-	Electrical steel AISI-M19C-4, copper wire MLT insulation
Bearing supports	-	Beryllium copper alloy 172
Bearings	-	440 C CRES
Bearing grease	-	Mobil 28
Static seals	-	Inconel, teflon coated.

2.7.2 Static Seals

Because of the critical importance of not having leaks in coolant systems for space vehicles having long duration missions, the selection of static seals is important. The approach selected was the use of teflon coated metallic o-rings clamped in the axial direction between flanges having substantial thickness (rigidity). This approach was used successfully in the ATM pump, 580745, and was used for the internal seals of the prototype pump, 581280. For test convenience only, the inlet and outlet flanges of the prototype pump, 581280, were designed for elastomeric seals. In a flight type design, these joints would use teflon coated metallic o-rings.

3. ATM PUMP DEMONSTRATOR PROGRAM

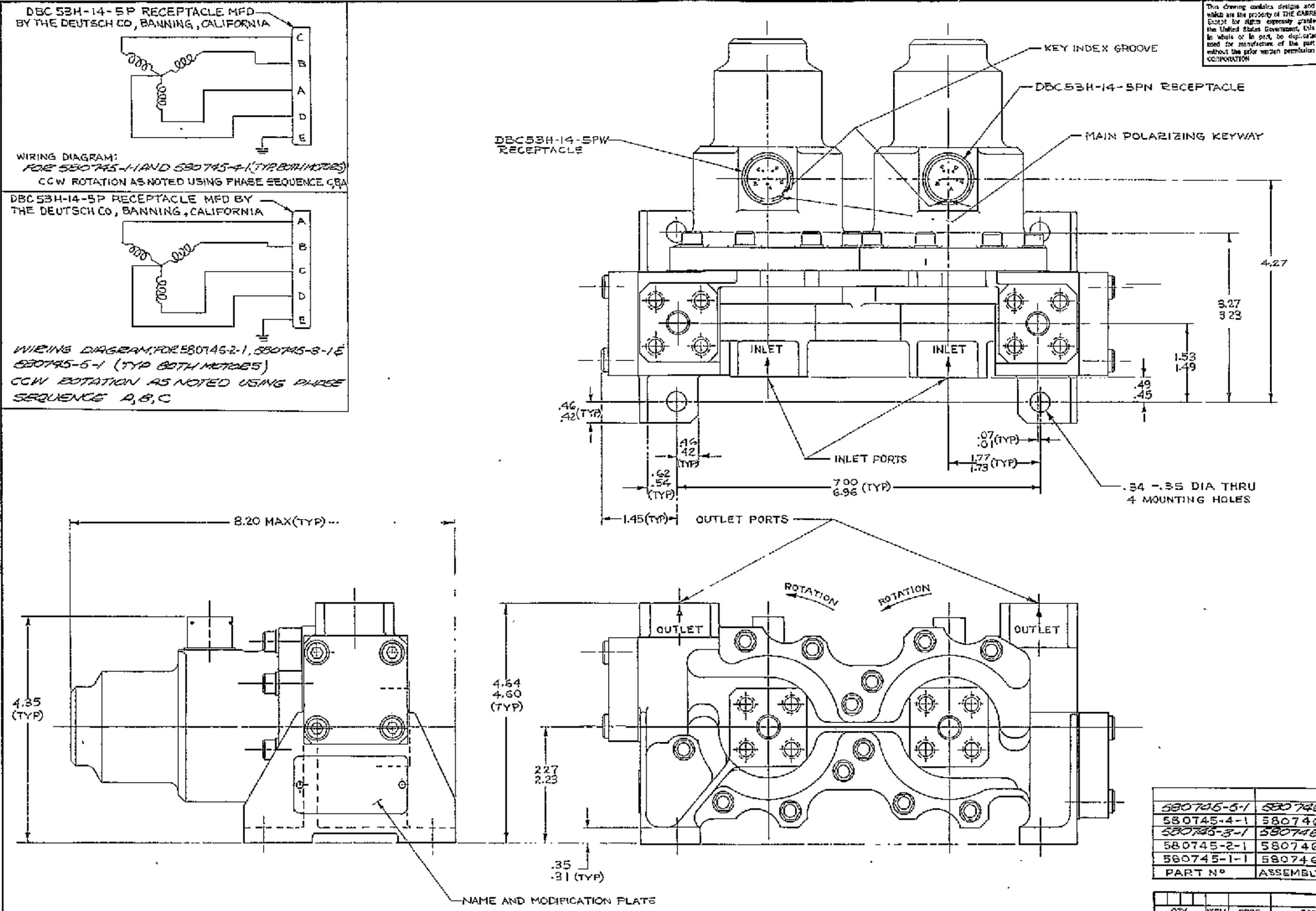
3.1 Pump Description

Figure 1-2 shows the external configuration of the ATM pump, 580745, and drawing 580745 shows the external dimensions. Figure 1-3 presents an exploded view of the ATM pump showing the basic design arrangement, including the magnetic coupling drive. The ATM pump used a mixture of water and methanol as the working fluid, and provided a flow of 900 PPH at a pressure rise of 31 PSID. The electrical input power was 3 phase, 12.2/21.2 VAC, 400 Hz, quasi-sine wave, and 125 watts were required per pump. Two pumps were included in a pumping package for redundancy. In service, one pump operated at a time, and reverse flow check valves were provided to prevent reverse flow in the non-operating pump.

The electrical motor was a dry motor, with the electrical rotor carried on grease packed ball bearings. A platinum cobalt magnetic coupling drive was used to eliminate the need for a dynamic shaft seal. The pump was a centrifugal type design with an unshrouded impeller. The pump rotating group was carried in a plain bearing made of Fiberite, which was a glass filled epoxy composite. During initial development, the Fiberite bearings posed a problem because of dimensional changes of the material after long term soak in the water-methanol mixture. Clearances were finally adjusted to provide reliable operation for the specified mission duration, but it was considered that this was an item which would require improvement for longer duration missions. Further details regarding the ATM pump development may be found in AiResearch Report No. 71-7254, Final Report, Methanol-Water Pump Package for ATM, P/N 580745, NASA Contract NAS8-30143, 10 June 1971.

3.2 Pump Design Modification

An initial task in the program was to establish a pump bearing concept which did not involve the use of materials subject to dimensional change in the working fluid, and to verify feasibility by test. For this purpose, an ATM pump, 580745, was modified to include a pressurized, double conical bearing. This bearing configuration used the double cone configuration to carry axial thrust loads applied from either direction, and to carry radial loads applied at either end of the pump impeller shaft. A schematic cross-section of the bearing is shown in Figure 2-6. Each conical bearing included four hydrostatic pads in the cone surface, and these were pressure fed with working fluid from a pressure tap located at the periphery of the impeller housing. Fluid pressure at this point was somewhat less than pump discharge pressure because the pressure tap was upstream of the pump conical diffuser, and some of the pump pressure rise is obtained by recovery in the diffuser. Pump working fluid which passed through the conical bearings was vented back to a low pressure area of the impeller housing by means of transfer ports. The male and female bearing cones were made of Inconel 706. The male cones were chrome plated to achieve a high surface hardness and the female cones were teflon coated to provide boundary



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REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED
A	SEE ENGINEERING ORDER	10-17-69	[illegible]
B	SEE ENGINEERING ORDER	2-3-70	[illegible]
C	SEE ENGINEERING ORDER	6-5-72	[illegible]
D	SEE ENGRG ORDER	6-16-72	[illegible]
E	SEE ENGRG ORDER	7-6-72	[illegible]
F	SEE E.O.	11-21-72	[illegible]

FOLDOUT FRAME 2

580745-5-1	580746-5	
580745-4-1	580746-4	
580745-3-1	580745-3	
580745-2-1	580746-2	SUPERSEDED BY 580745-5-1
580745-1-1	580746-1	SUPERSEDED BY 580745-2-1
PART Nº	ASSEMBLY Nº	REMARKS

QTY REQD	ITEM NO.	CODE IDENT NO.	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	SYM
	← ASSY			PARTS LIST	

5. PUMP: (TYP)
FLUID - 80% METHANOL, 20% WATER, FLOW - 900 LBS/HR, PRESSURE RISE - 31 PSID
4. MOTOR: (TYP)
12.2 VAC RMS, 400 H_Z, 3 PHASE, 0.1 HP
3. INLET AND OUTLET PORTS PER MSFC DWG 20M42522-1
2. ALL OPENINGS COVERED FOR SHIPPING AND STORAGE PURPOSES ONLY. REMOVE AT TIME OF INSTALLATION
1. ALL DIMENSIONS FOR INSTALLATION PURPOSES ONLY. TOLERANCES ±.12
NOTES: UNLESS OTHERWISE SPECIFIED

FOLDOUT FRAME 1

	FINAL	
REQD	NEXT ASSY	USED ON
APPLICATION		

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES.
MACHINE FILLET RADII .015 — .030
SURFACE ROUGHNESS PER USAS B46.1-62
BURR CONTROL PER SC553
DIMENSION LIMITS HELD AFTER PLATING
~~MARKING PER MS~~
DIMENSIONING AND TOL PER USAS Y14.5-66

HEAT TREATMENT	PROCESS
HARDNESS AND SPEC	NAME AND SPEC
—	—

CONTRACT NO.
DFT L.W. Jackson 4-29-69
CHK [illegible] 9-5-69
VALUE ENGR
MATL
STRESS
APPD [illegible] [illegible]
APPD [illegible] 4-30-69
AIRESEARCH APPD [illegible] 9/25/69
OTHER ACTIVITY APPD

AIRESEARCH MANUFACTURING COMPANY
A DIVISION OF THE GARRETT CORPORATION
LOS ANGELES, CALIFORNIA

PUMP UNIT OUTLINE, MOTOR DRIVEN

SIZE D | CODE IDENT NO. 70210 | DWG NO. 580745
SCALE 1/2 | MAX WT 10.35 LB | SHEET OF

580745

lubrication during starts and stops. Appendix B presents a first order design analysis for this bearing.

Figure 3-1 shows an exploded view of the double conical bearing for the pump rotating group. Figure 3-2 shows the assembled pump rotating group. Figure 3-3 shows the assembled pump cartridge including the double conical bearings, and Figure 3-4 shows how the pump cartridge fits into the overall pump assembly.

The weight of the modified ATM pump, 580745, was 11.0 pounds.

3.3 Test Program

3.3.1 Initial Shakedown Tests

Initial tests were performed with water as the working fluid for test convenience. The first tests were static bench tests designed to make a first order determination of the bearing optimum assembly clearance and pressurant flow. Axial clearances from 0.0005 inches total to 0.0020 inches total, and bearing pressure differentials from 40 psid down to 5 psid were investigated.

Based upon analysis and upon the bench test results, an axial clearance of 0.0017 inches total was selected and runs performed with the complete pump. Figure 3-5 shows a photograph of the water test loop and related instrumentation. Initial runs were performed with the bearing pressurized by an external source. Later tests were performed with the pressurant obtained from a tap at the pump discharge, i.e., the system was self-pressurizing. Pressure differential across the bearings was controlled by an adjustable valve. A series of careful tests and disassembly inspections showed that the bearing pressurant flow could be reduced to 2 percent of the pump through flow, with acceptable bearing performance. Approximately 10 hours run time and 30 starts were accumulated with no visible discrepancies on the bearing surfaces. While operation was satisfactory, it was desired to increase the bearing compensation, and this was done by installing restrictor orifices in the pressurant feed lines to the individual hydrostatic pockets. At the conclusion of the water tests, check-out runs were made satisfactorily, using Freon 21 as the working fluid. Figure 3-6 shows a photograph of the Freon 21 test loop and related instrumentation.

3.3.2 Calibration Test

In the initial shakedown tests, satisfactory pump operation was established using Freon 21 as the working fluid. A pump calibration was then performed, and the results of the calibration are shown plotted in Figure 3-7. This is the performance which was achieved from an ATM pump, 580745, modified to include double conical bearings for the pump rotating group, and operated with Freon 21 as the working fluid.

3.3.3 Endurance Test

After calibration, the pump was put on long term endurance test, to determine whether the performance and operation of the pump bearing would

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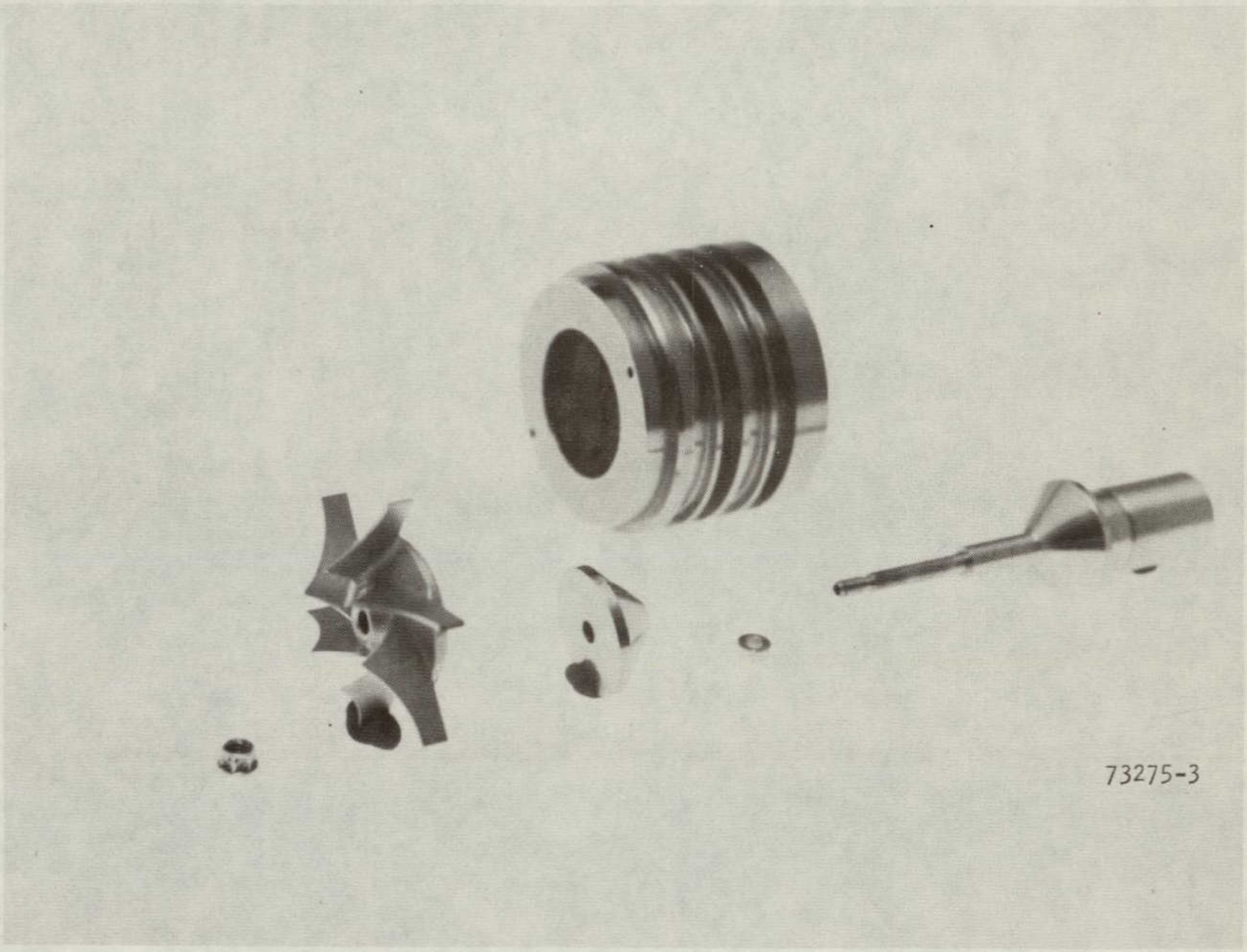

Figure 3-1. Exploded View of Double Conical Bearing for Pump Rotating Group of Modified ATM Pump, 580745.

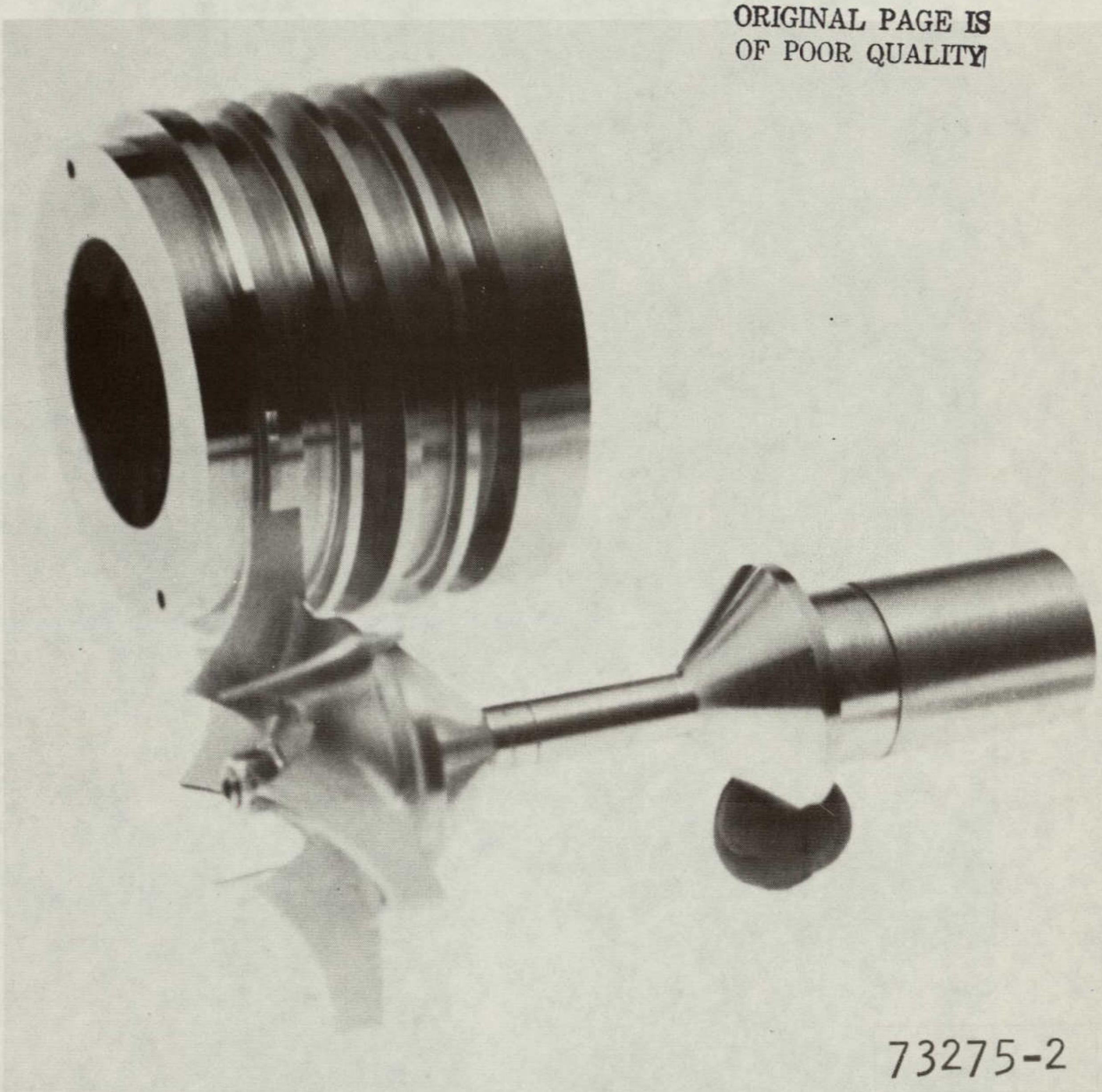

Figure 3-2. Assembled Pump Rotating Group
for Modified ATM Pump, 580745

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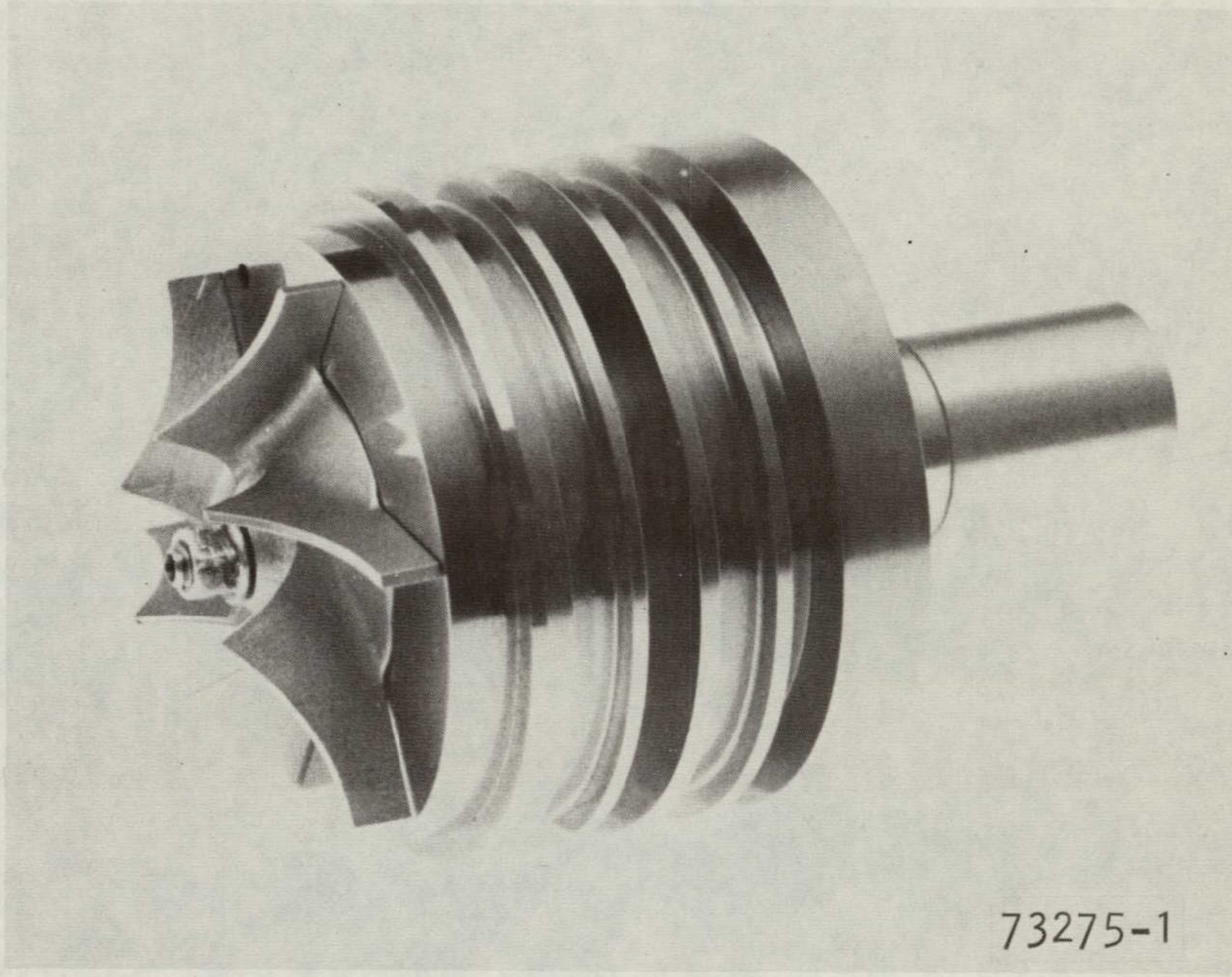

Figure 3-3. Assembled Pump Cartridge for
Modified ATM Pump, 580745.

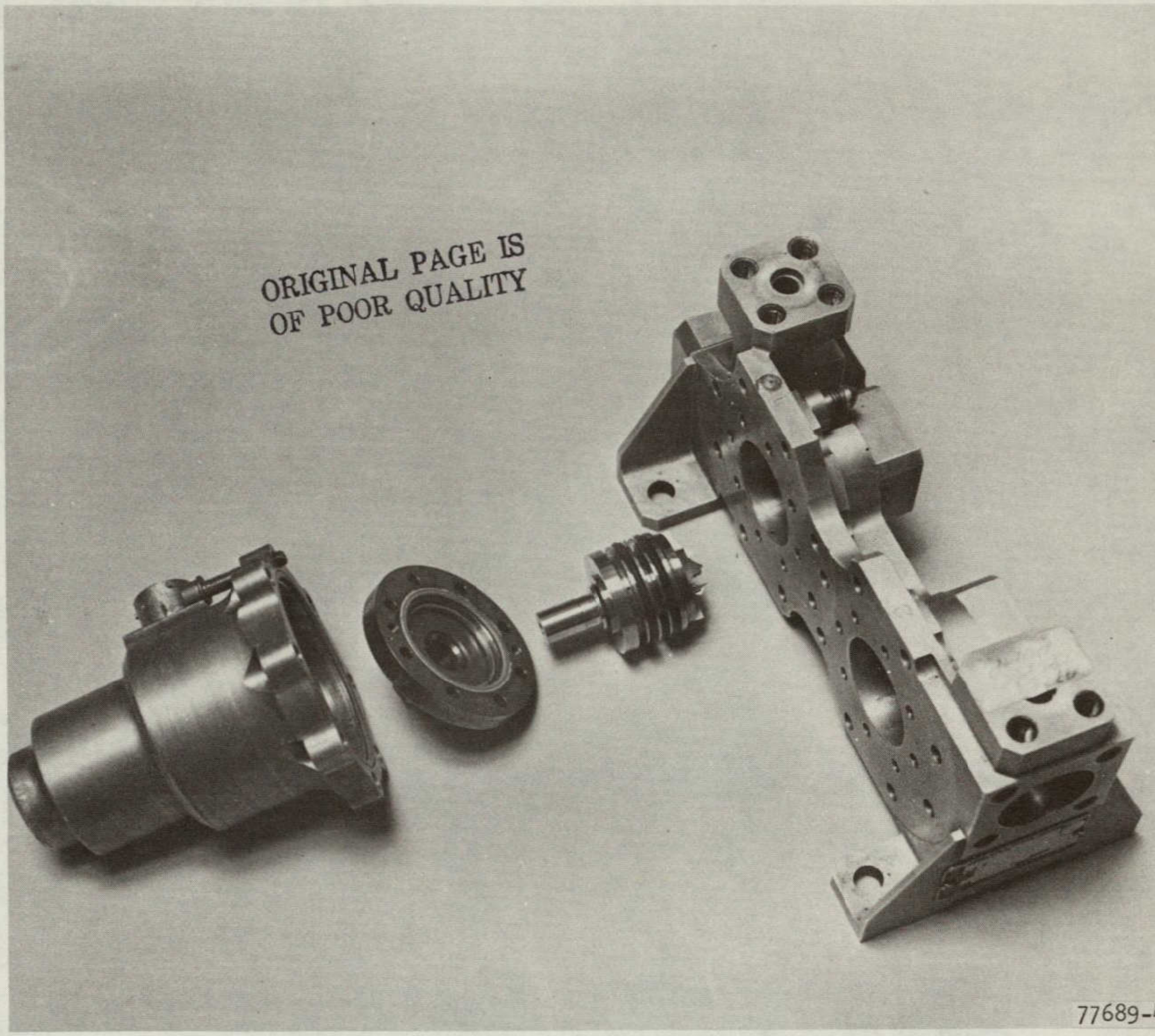

Figure 3-4. Exploded View of Modified ATM Pump, 580745, Including Pump Cartridge with Double Conical Bearing.

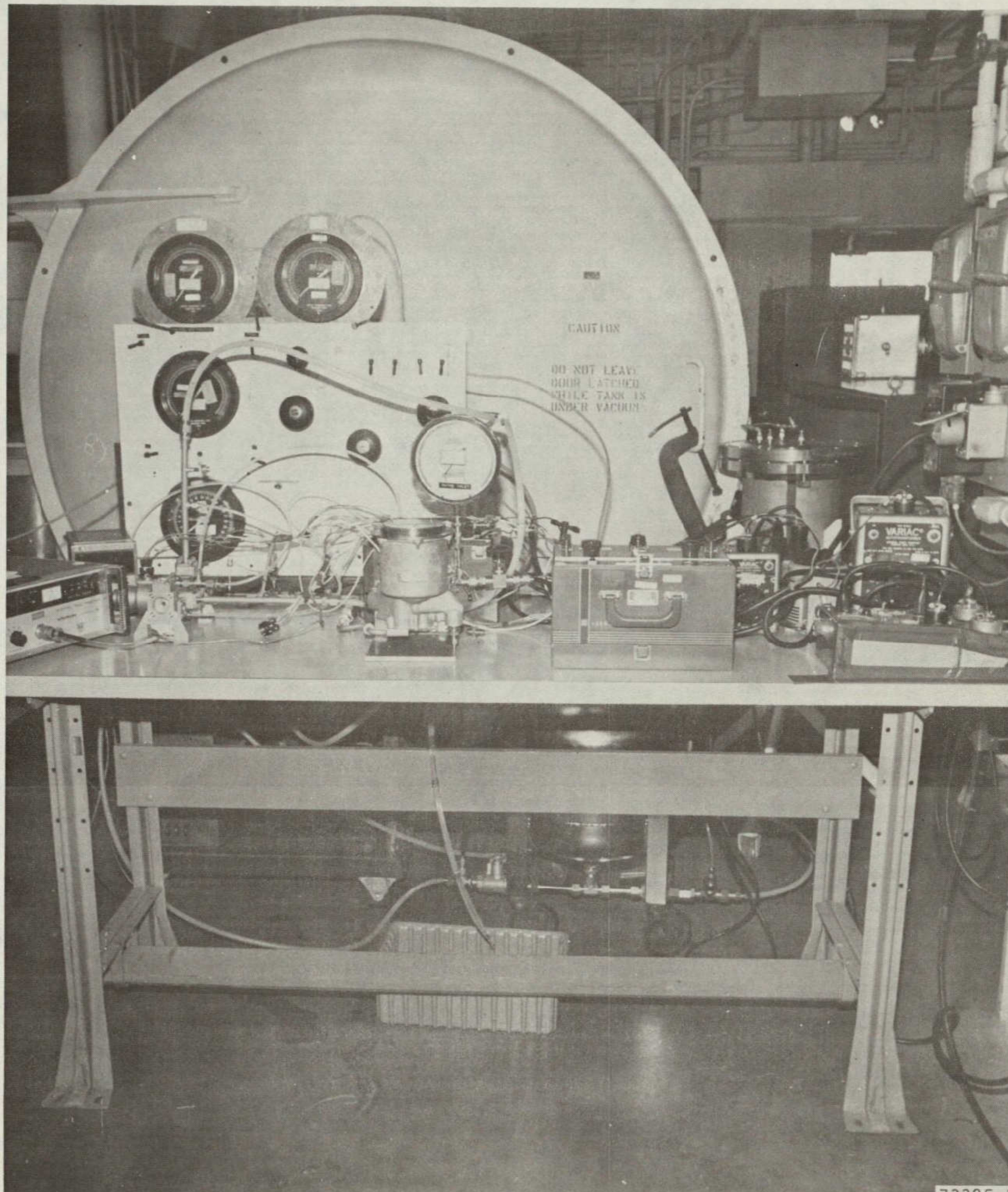

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Figure 3-5. Showing Water Test Loop, and Related Instrumentation.

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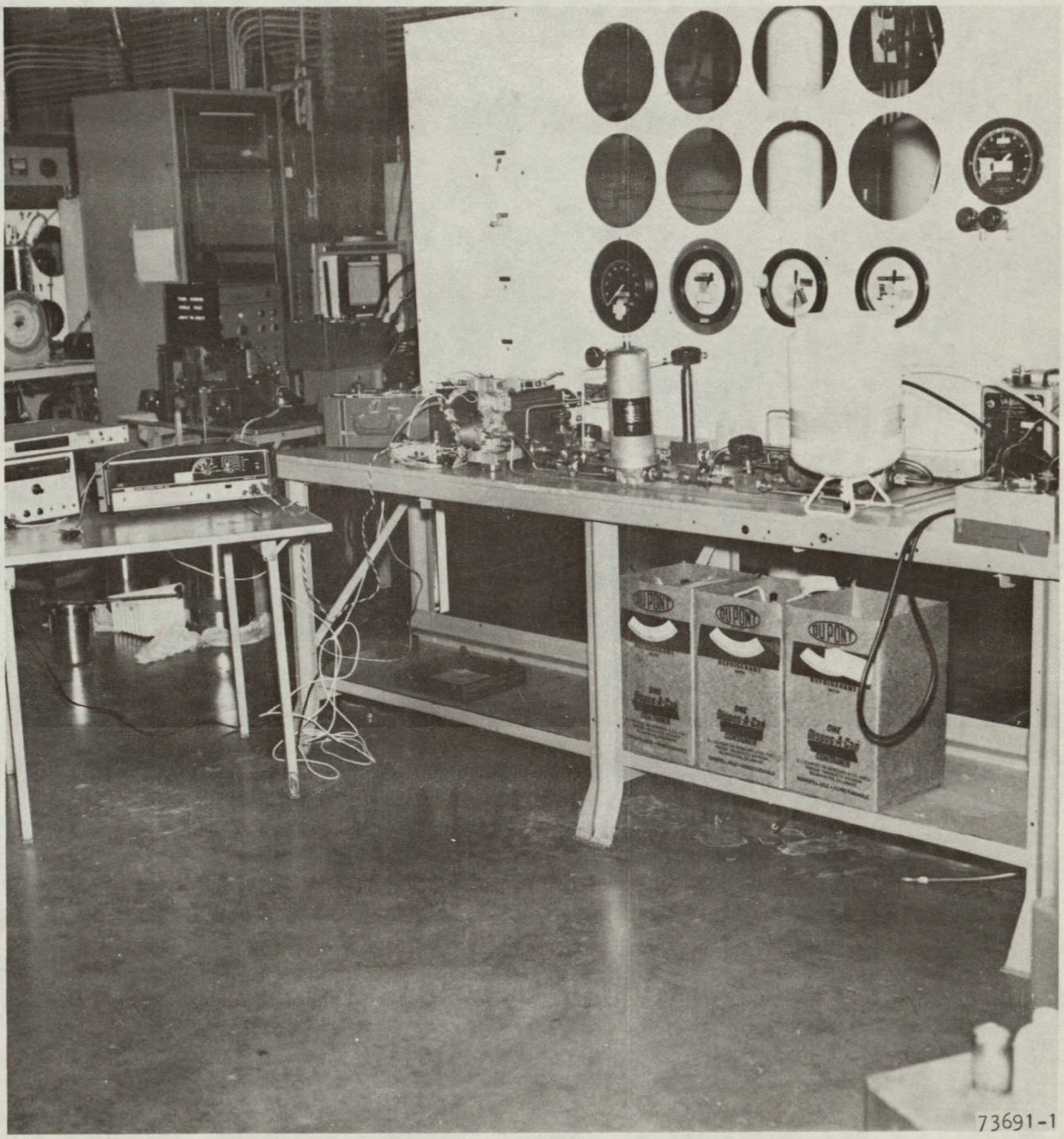

73691-1

Figure 3-6. Showing Freon 21 Test Loop, and Related Instrumentation.

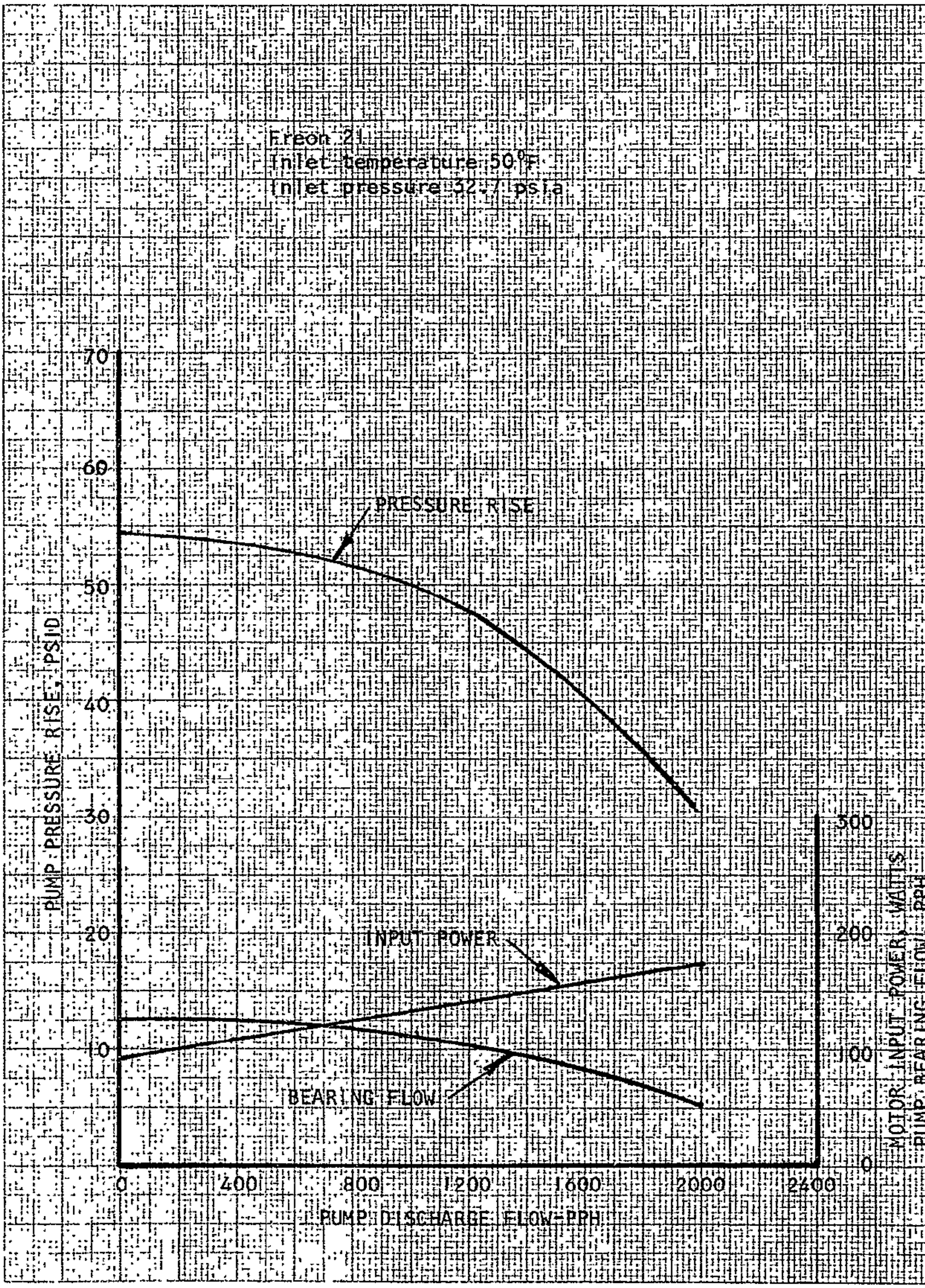

Figure 3-7. Performance of Modified ATM Pump, 580745, Early in Test Program.

change significantly with time. This was essentially an unattended test, with pump hydrodynamic and electrical parameters logged once a day, until the termination of the program. A total endurance time of 12,304 hours and over 160 starts and stops were accumulated during this running, with no significant change in observed operating parameters, except for bearing flow.

3.3.4 Post Endurance Calibration Test

After termination of the endurance running, a post endurance calibration test was performed, and results of the calibration are shown plotted in Figure 3-8. Comparison of Figure 3-8 with Figure 3-7 shows very little change in pump performance as a result of the endurance test.

3.3.5 Disassembly Inspection

After the post-endurance calibration test, the pump was disassembled for inspection, and in general the findings were considered good.

Figure 3-9 shows the front of the impeller and Figure 3-19 shows the rear of the impeller after the endurance test, with no significant rub marks.

Figure 3-11 shows the impeller end bearing cone. This has several circumferential contact marks, which were believed to occur during the starts and stops when pressurant flow is not available. The marks did not appear to affect the bearing operation. A profilometer measurement showed the finish to be RMS 14, compared to the initial value of RMS 8.

Figure 3-12 shows the magnet end bearing cone, and this also had some circumferential contact marks. A profilometer measurement showed the finish to be RMS 20, compared to the initial value of RMS 8.

Figure 3-13 shows the impeller and female bearing cone. Some circumferential contact marks are evident, and this is primarily a wearing away of the teflon coating. A profilometer measurement showed the finish to be an average of RMS 20, except for the grooves, and this compared to the initial RMS 8.

Figure 3-14 shows the magnet and female bearing cone. Some circumferential contact marks are evident, and this is primarily a wearing away of the teflon coating. A profilometer measurement showed the finish to be an average of RMS 20, except for the grooves, and this compared to the initial RMS 8.

Figure 3-15 shows the motor bearings after the endurance test. The bearings were in excellent condition as described by the bearing disassembly inspection report shown in Figure 3-16. The bearings showed very little change after 12,304 hours, and it is estimated that the bearings could operate another 10,000 hours or more, judging by the excellent condition of the grease.

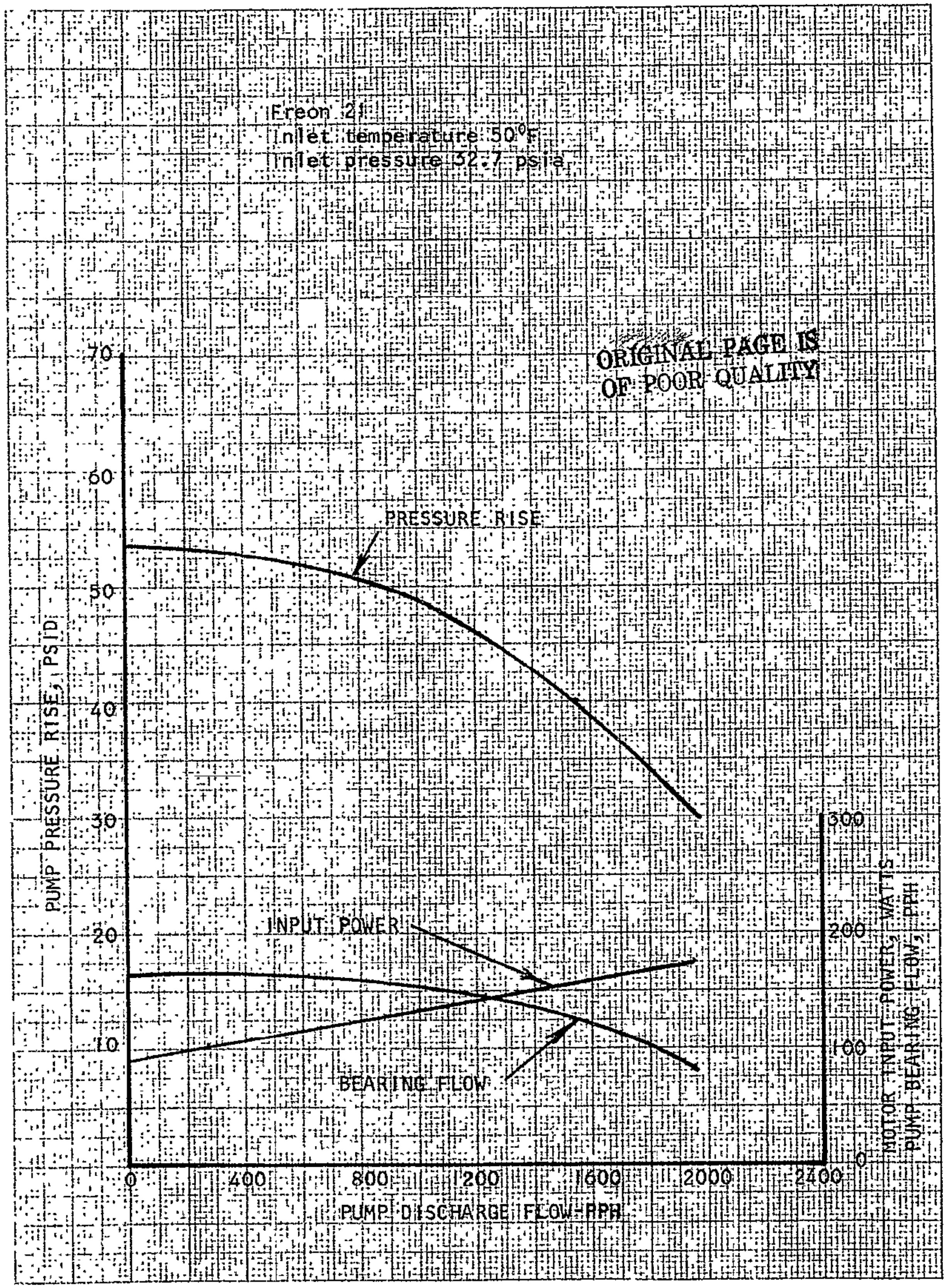

Figure 3-8. Performance of Modified ATM Pump, 580745, at Completion of Test Program

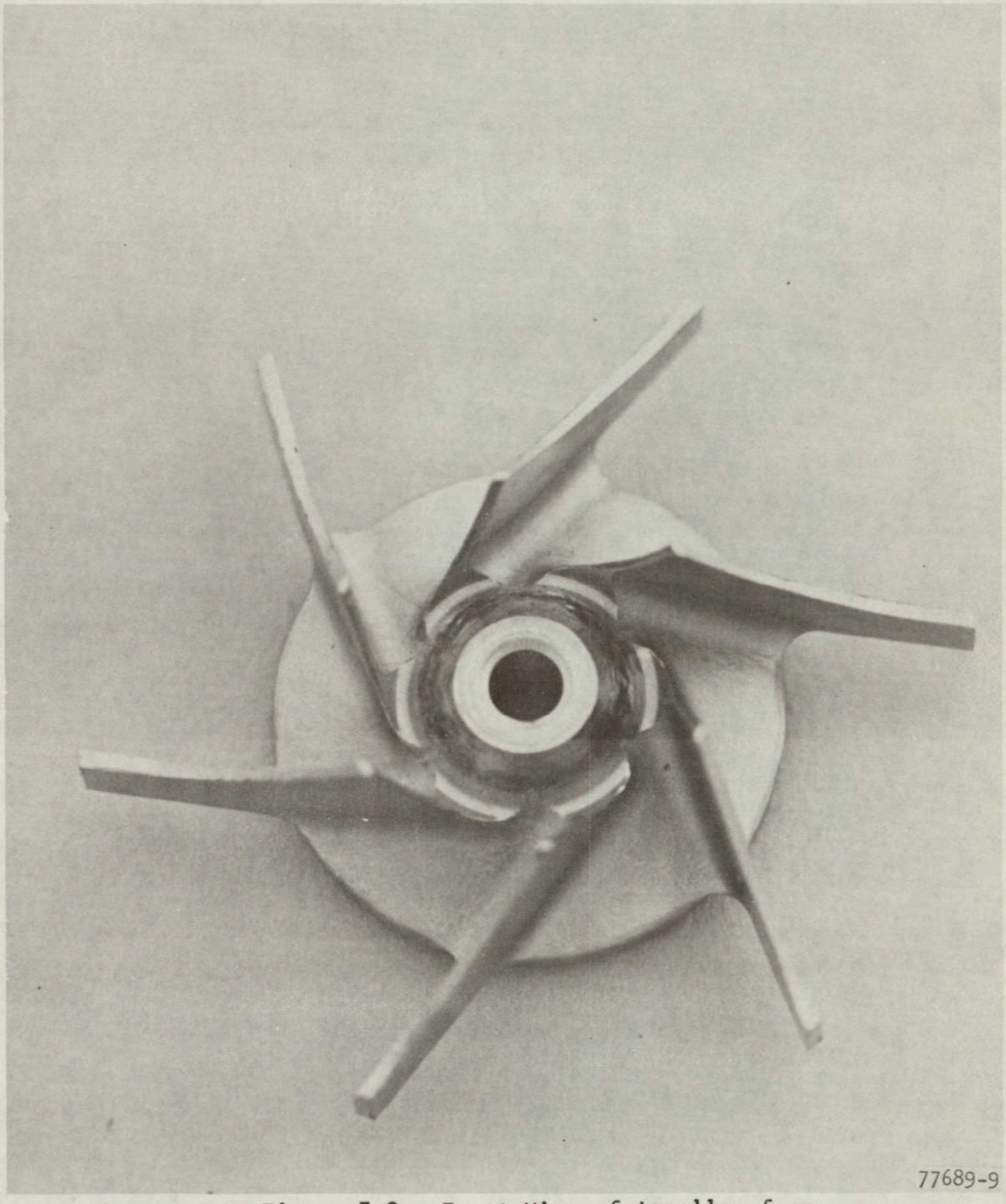

Figure 3-9. Front View of Impeller from Modified ATM Pump, 580745, after 12,304 Endurance Hours.

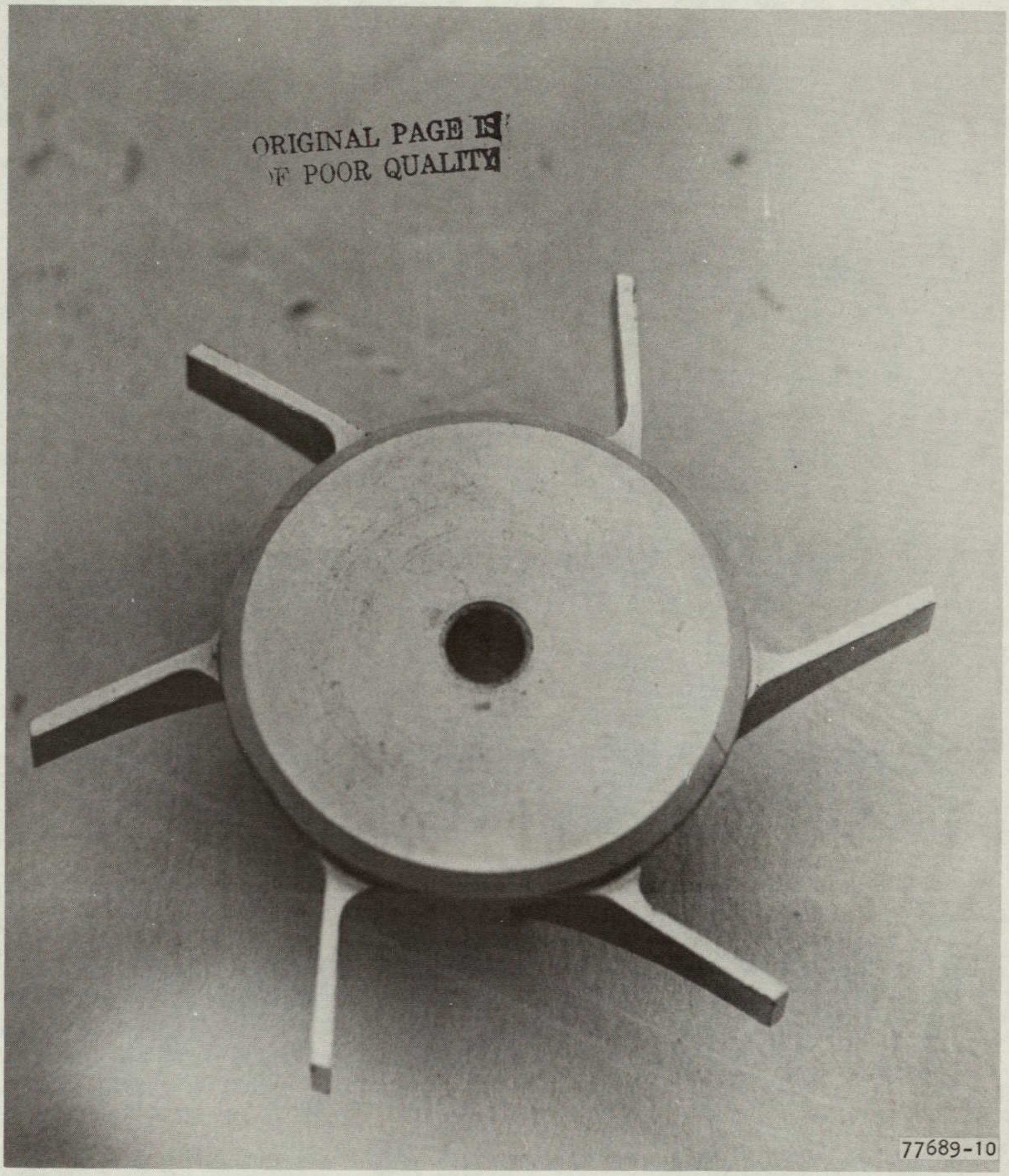

Figure 3-10. Rear View of Impeller From Modified ATM Pump, 580745, After 12,304 Endurance Hours.

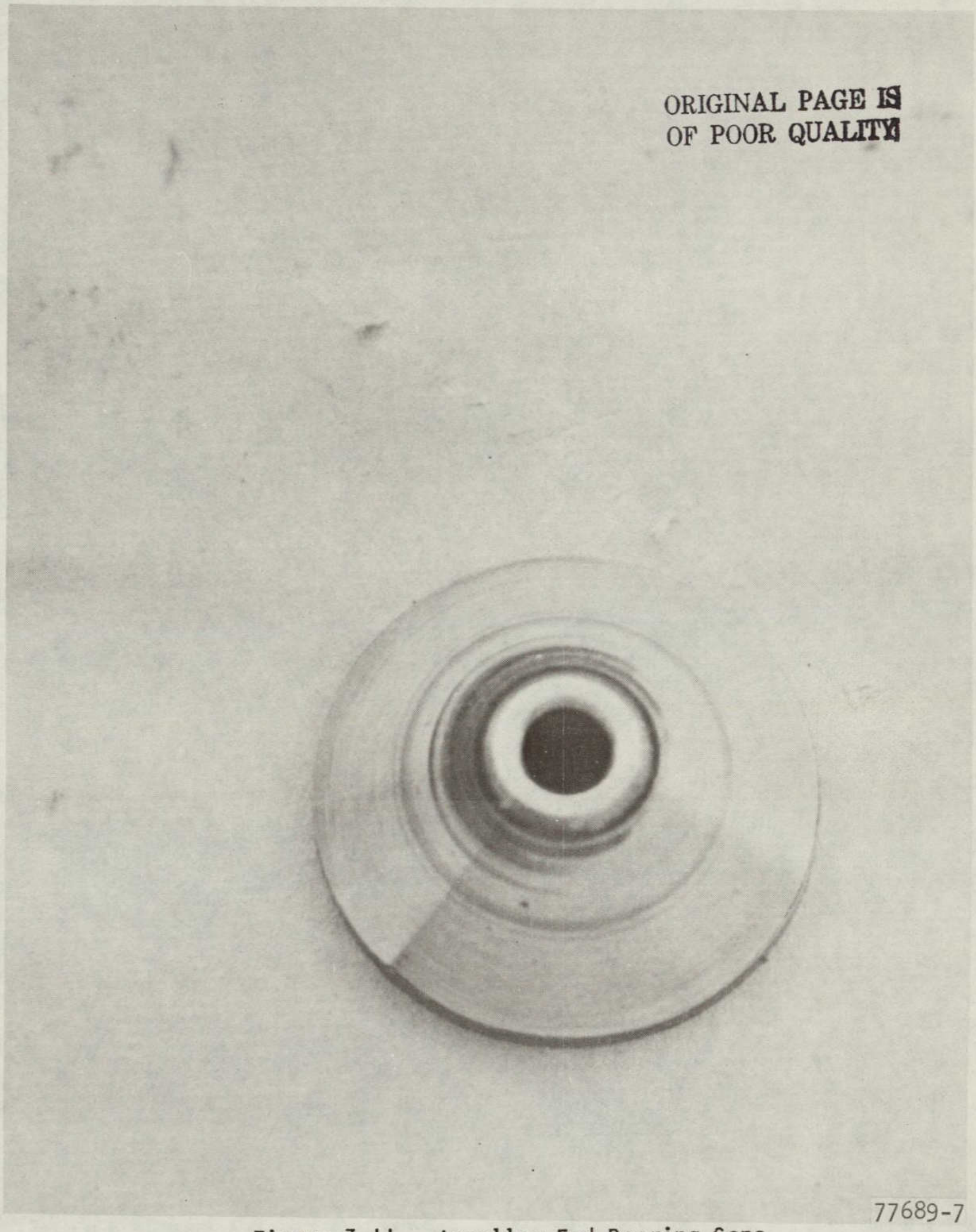

Figure 3-11. Impeller End Bearing Cone
From Modified ATM Pump, 580745,
After 12,304 Endurance Hours.

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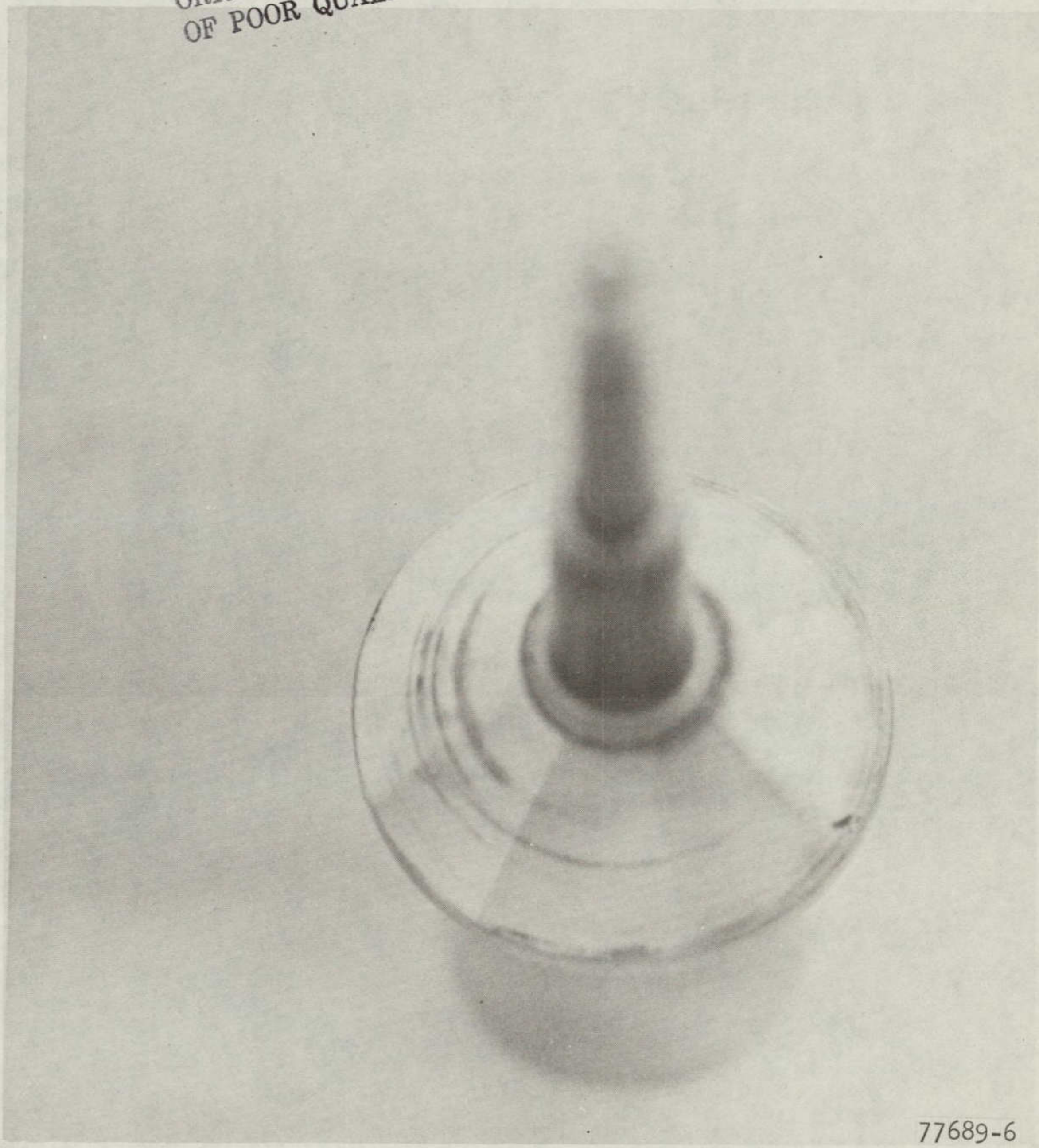

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Figure 3-12. Magnet End Bearing Cone
From Modified ATM Pump, 580745,
After 12,304 Endurance Hours.

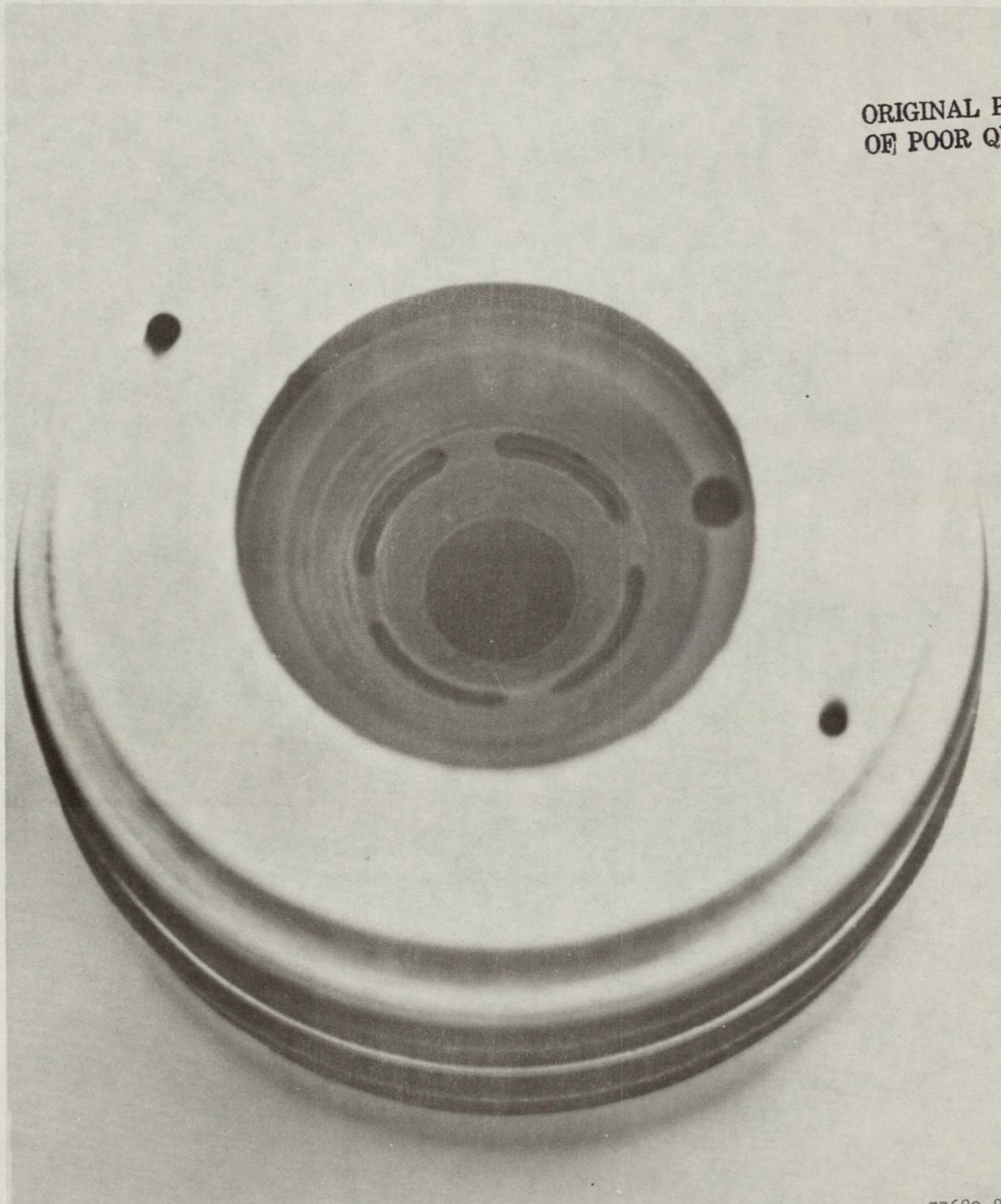

Figure 3-13. Impeller End Female Bearing Cone From Modified ATM Pump, 580145, After 12,304 Endurance Hours.

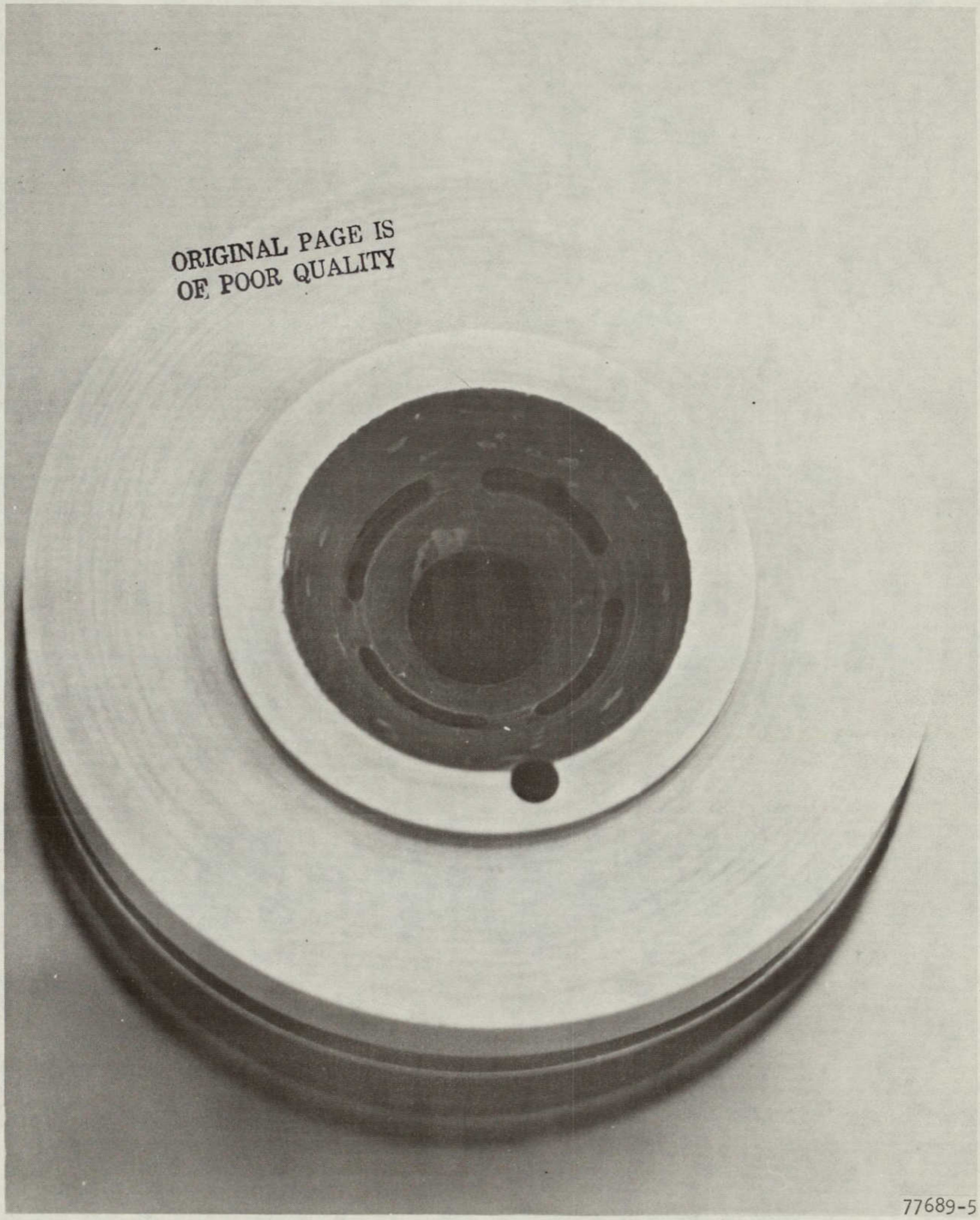

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Figure 3-14. Magnet End Female Bearing Cone
From Modified ATM Pump, 580745,
After 12,304 Endurance Hours.

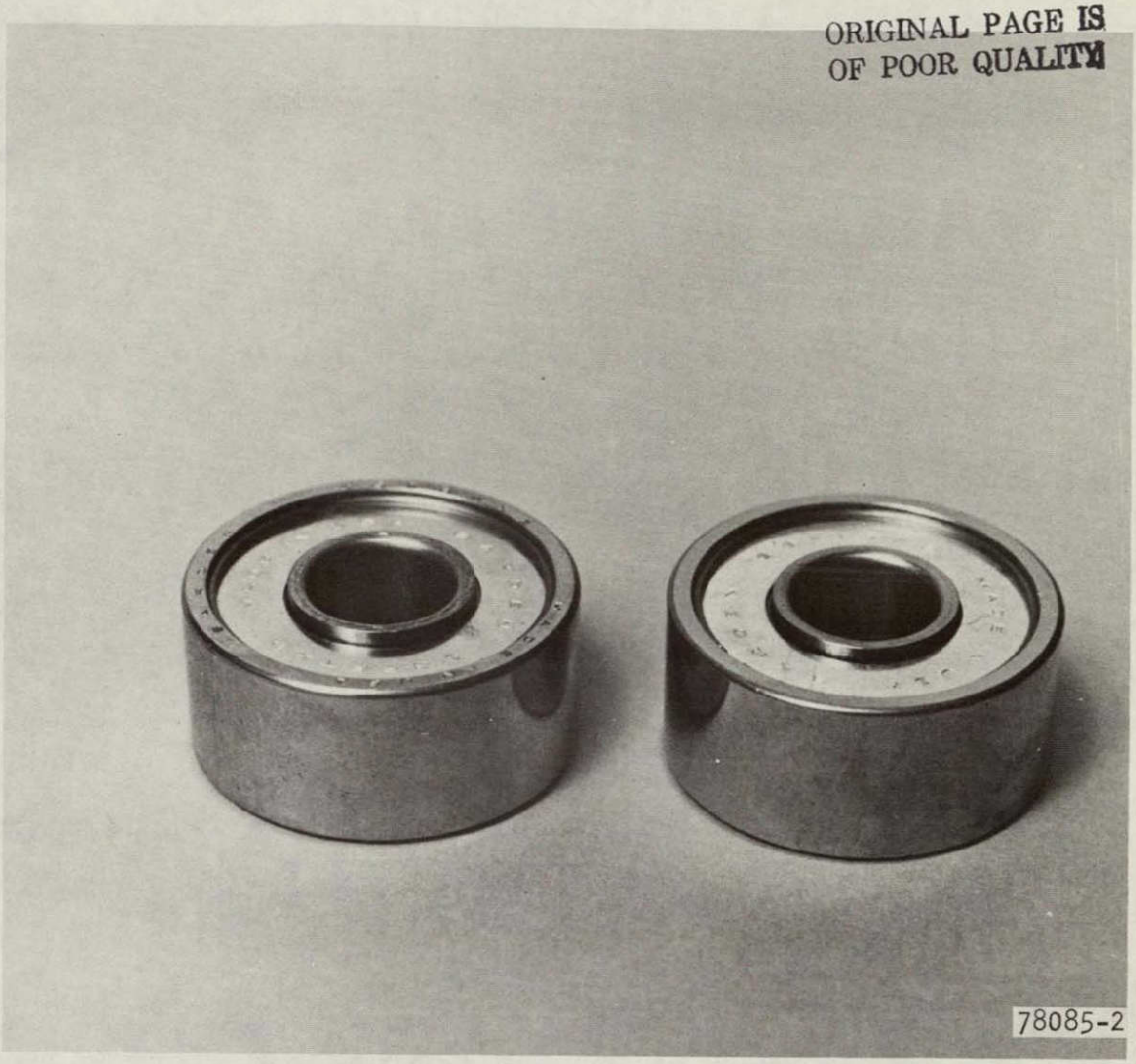


Figure 3-15. Motor Bearings, 580629-1, After 12,304 Hours of Operation in Modified ATM Pump, 580745.

OPERATING DATA

12304 HOURS

BEARING REPORT BR 770308-8-1622

UNIT

P/N 580745

S/N

BEARING BARDEN

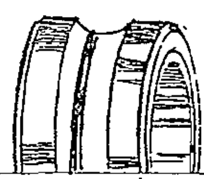
DIST. J. RIPLE

R.O. 300-08-77-8864

NO. I MAGNET END BEARING .176 GRAM GREASE IN THE BEARING.

INNER RING

BRIGHT SURFACES. FAINT LIGHT LOAD TRACK.

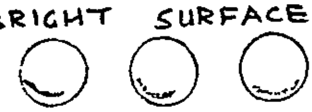


SEPARATOR

GLAZED OD AND POCKET SURFACE

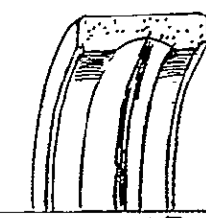
BALLS

BRIGHT SURFACES.



OUTER RING

OD DID NOT CREEP IN THE HOUSING. BRIGHT SURFACES.

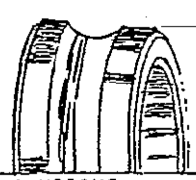


FAINT LIGHT LOAD TRACK.

NO. II REAR END BEARING .162 GRAM GREASE IN THE BEARING.

INNER RING

BRIGHT SURFACES. FAINT LIGHT LOAD TRACK.

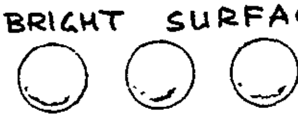


SEPARATOR

GLAZED OD AND POCKET SURFACE.

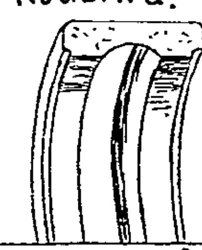
BALLS

BRIGHT SURFACES.



OUTER RING

SUPERFICIAL OD FRET IN THE HOUSING. BRIGHT SURFAC



FAINT LIGHT LOAD TRACK.

CONCLUSIONS: THE BEARINGS ARE IN EXCELLENT CONDITION. THE RACEWAY, BALL AND SEPARATOR SURFACES DO NOT EXHIBIT WEAR, HEAVY LOADING OR ADVERSE OPERATING CONDITIONS.

THE GREASE IN THE BEARINGS IS EVENLY DISTRIBUTED, SMEARS READILY AND SHOWS NO SIGNS OF IMPENDING BREAKDOWN.

THE BEARINGS SEEM TO BE AS GOOD AS NEW AND WOULD PROBABLY OPERATE FOR 10000 MORE HOURS OR MORE.

CONDITION	I	II
BALANCE		
LOADS		
ALIGNMENT		
LUBRICATION		

3-8-77
DATE

R. Bhikha
BEARING COORDINATOR

Figure 3-16. Results of Inspection of Bearings from Modified ATM Pump, 580745, after 12,304 Hours of Operation.

4. PROTOTYPE PUMP DEMONSTRATOR PROGRAM

4.1 Pump Description

Figure 4-1 shows the external configuration of the prototype pump, 581280, and drawing 581280 shows the external dimensions. Drawing SK 65924 shows the pump and motor assembly, and the basic design arrangement which is approximately the same as that of the modified ATM pump, 580745. It should be noted that the prototype pump, 581280, was a single pump and not a dual pump package as the ATM pump, 580745.

Figure 4-2 presents an exploded view of the prototype pump, showing the major sub-assemblies.

The pump operated with Freon 21 working fluid and had a flow of 2200 PPH at a pressure rise of 55 psid. The electrical input power was 3 phase, 115/200 VAC, 400 Hz, sine wave, and 300 watts were required at the design point.

Figure 4-3 shows a cross-sectional drawing of the prototype pump. The electrical motor was a dry motor, with the electrical rotor carried on grease packed ball bearings. A coolant passage was provided in the motor stator housing to facilitate cooling of the motor. This proved to be unnecessary at the ambient conditions used in the test program, but would be desirable for certain type applications. A platinum cobalt magnetic coupling drive was used to eliminate the need for a dynamic shaft seal. The pump was a centrifugal type design with an unshrouded impeller. The pump rotating group was carried on pressurized double conical bearings which could carry axial thrust loads applied from either direction, and radial loads applied at either end of the pump impeller-shaft. Each conical bearing included four hydrostatic pads in the cone surface, and these were pressure fed with working fluid from a pressure tap located at the periphery of the impeller housing. Fluid pressure at this point was somewhat less than pump discharge pressure because the pressure tap was upstream of the pump conical diffuser, and some of the pump pressure rise was obtained by recovery in the diffuser. Pump working fluid which passed through the conical bearings was vented back to a low pressure area of the impeller housing by means of transfer ports. The male and female bearing cones were made of Inconel 706. The male cones were chrome plated to achieve a high surface hardness and the female cones were teflon coated to provide boundary lubrication during starts and stops.

Figure 4-4 shows the hydrodynamic design configuration of the prototype pump.

Figure 4-5 shows the calculated performance of the motor for the prototype pump. The prototype pump motor preliminary thermal analysis is presented in Appendix A.

The weight of the prototype pump, 581280, was 6.9 pounds. It was estimated that a flight weight version would weigh 4.8 pounds.

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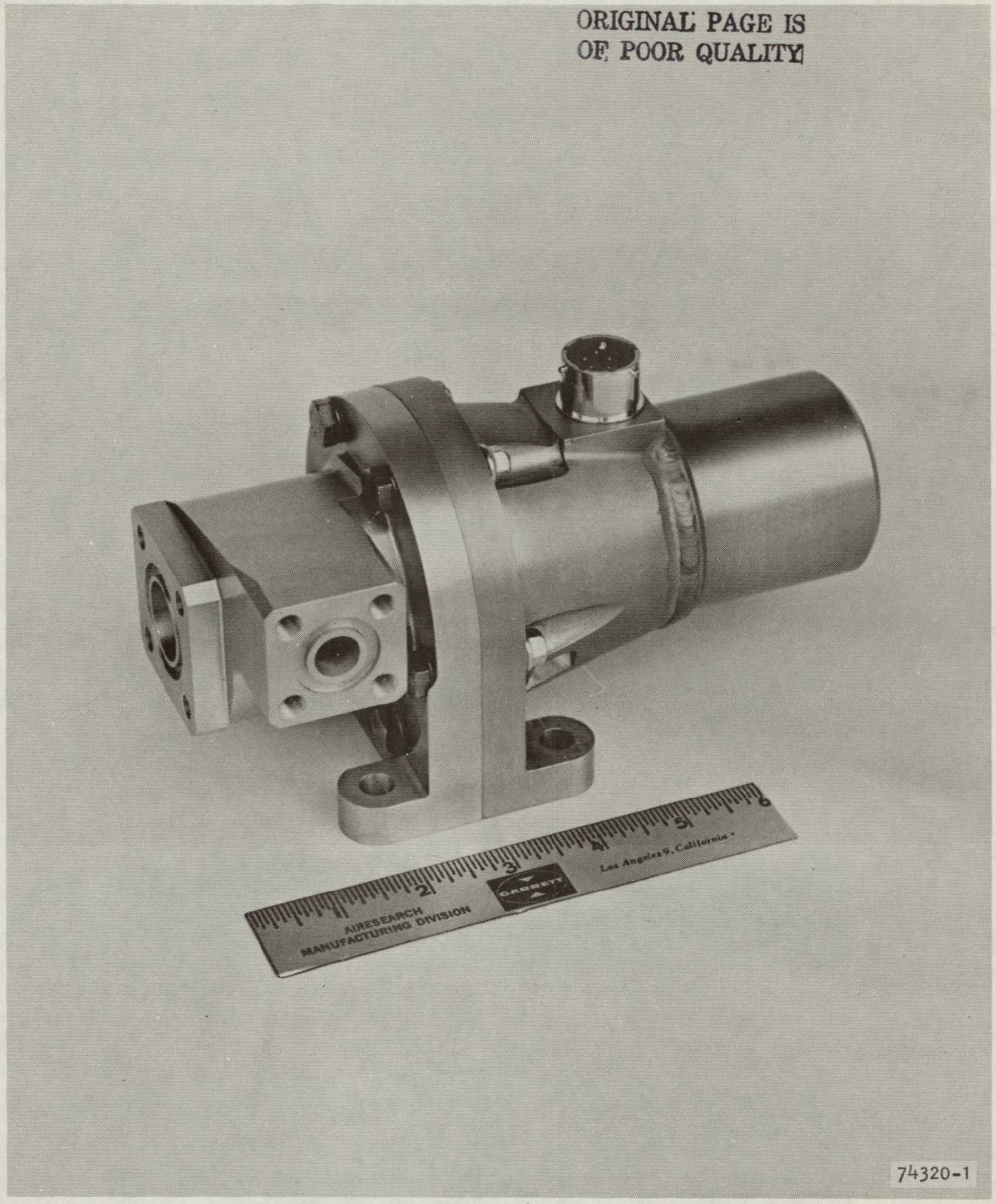
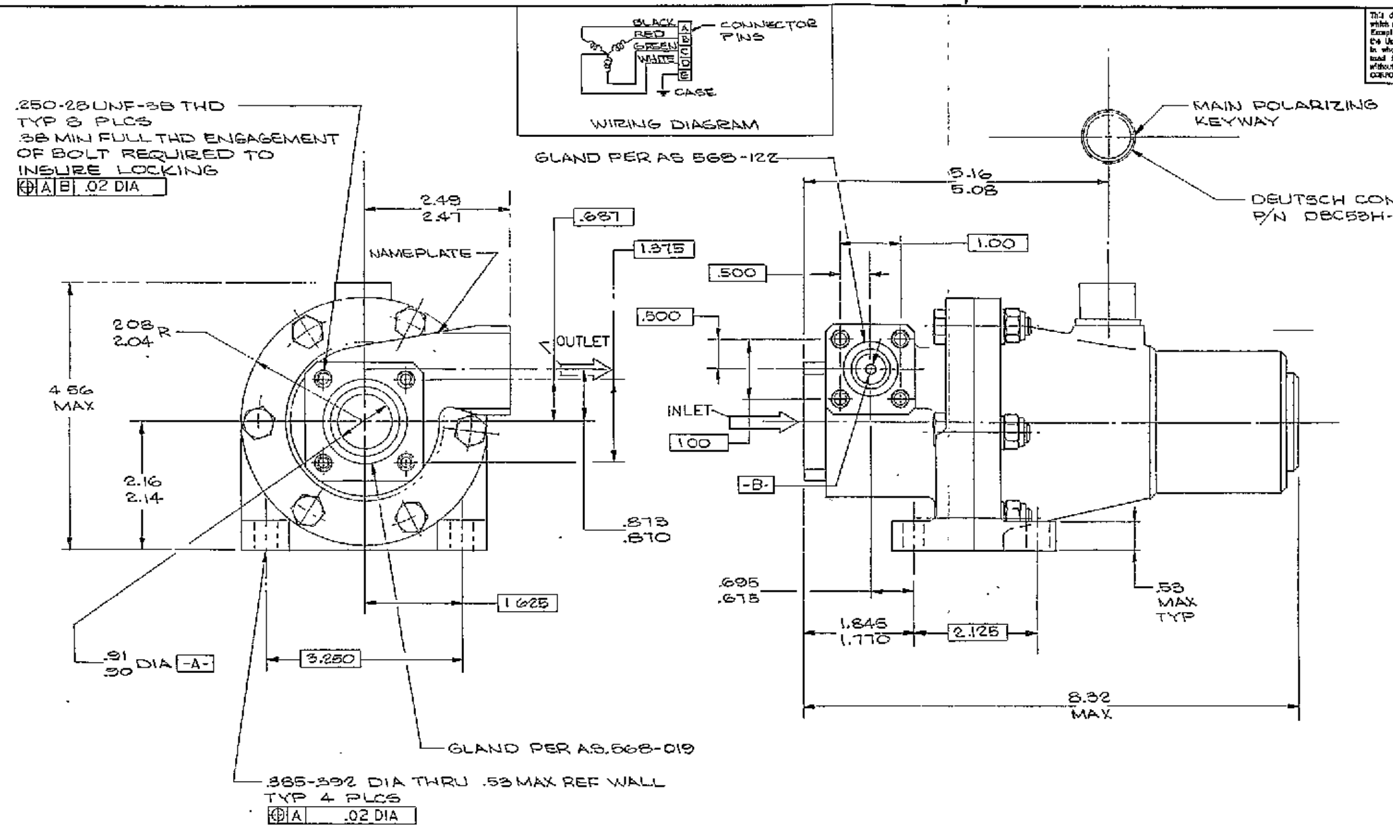


Figure 4-1. Prototype Pump, 581280

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED



FOLDOUT FRAME 1

FOLDOUT FRAME 2

2 ELECTRICAL INPUT 115/200 V, 400 HZ, 3 PHASE, 1.5 AMP/PHASE AT DESIGN LOAD OF 0.32 MOTOR HP
1 ALL DIMENSIONS FOR INSTALLATION PURPOSES ONLY
NOTES: UNLESS OTHERWISE SPECIFIED

PART NO	ASSY NO
581280-1-1	

QTY REQD	ITEM NO	CODE IDENT NO	PART OR IDENTIFYING NO	NOMENCLATURE OR DESCRIPTION	SYM
← ASSY				PARTS LIST	

FINAL		
REQD	NEXT ASSY	USED ON
APPLICATION		

UNLESS OTHERWISE SPECIFIED: BURR CONTROL PER SC653
IDENTIFICATION MARKING PER MC16
STD INTERPRETATIONS PER PIBS

HEAT TREATMENT — HARDNESS AND SPEC
PROCESS — NAME AND SPEC

CONTRACT NO
DFT J. Johnson 5-22-74
CHK
APPD 5-23-74
AIRESEARCH APPD 5-23-74
OTHER ACTIVITY APPD

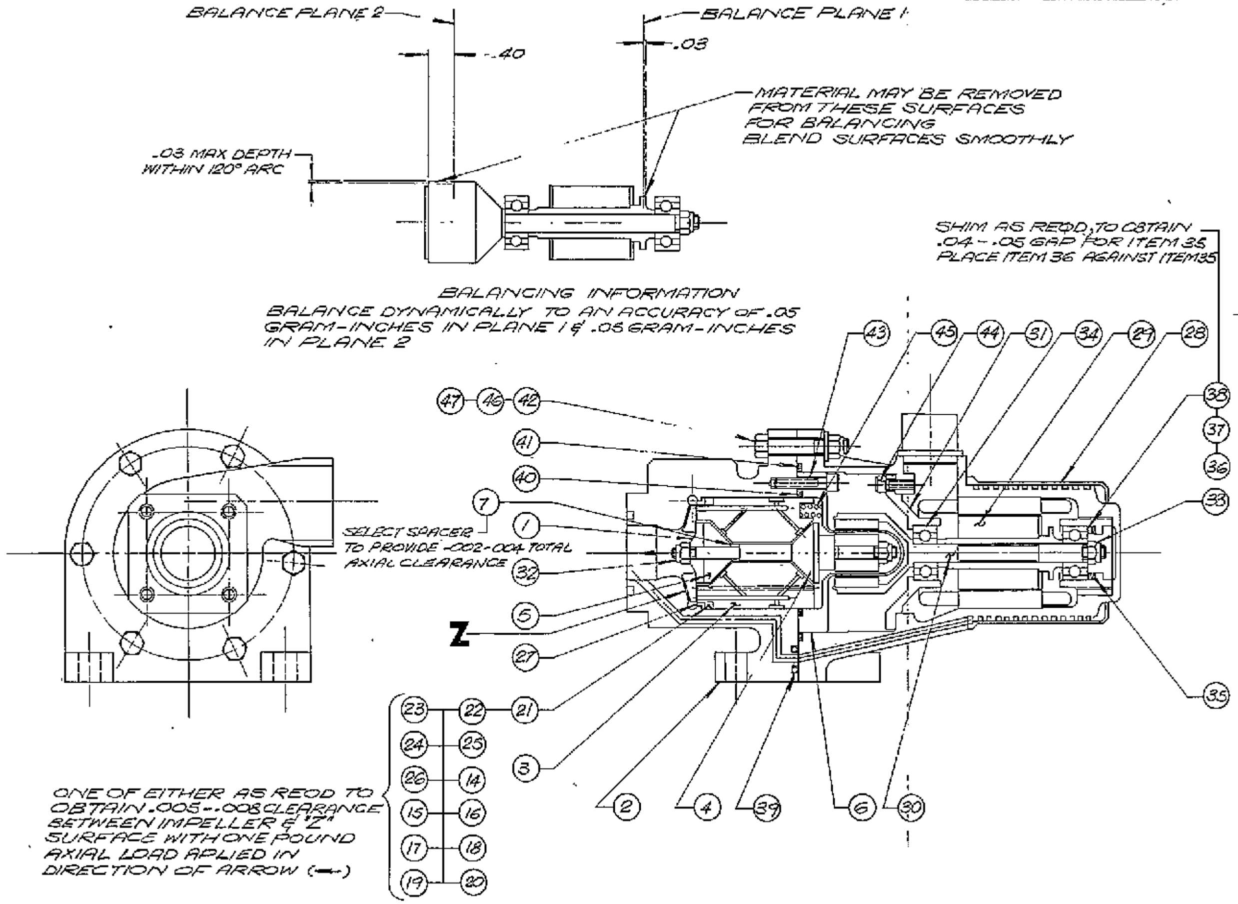
AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA
A DIVISION OF THE GARRETT CORPORATION
TORRANCE, CALIFORNIA

PUMP OUTLINE, COOLANT

SIZE	CODE IDENT NO	DWG NO
D	70210	581280

SCALE 1/1 SHEET 1 OF 1

581280



FOLDOUT FRAME 1

FOLDOUT FRAME 2

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REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED

QTY REQD	ITEM NO.	CODE IDENT NO	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	SYM
6	47		NAS1291C5	NUT	
6	46		MS9491-18	SCREW 5/16-24	
4	45		MS24585C128	SPRING	
4	44		MS16995-26	SCREW #8-32	
8	43		655-590-9602	SCREW #10-32	
6	42		S8154-50	WASHER	
1	41		3000-5-3NP2030PF	METALIC "O" RING	⚠1
1	40		2000-3-3NP2030PF	METALIC "O" RING	⚠1
2	39		S9026D11	"O" RING	
AR	38		S8157N106-005	WASHER	
AR	37		" -010	"	
AR	36		S8157N106-025	WASHER	
1	35		S8809A68	WASHER, SPRING	
2	34		580629-1	BEARING	
1	33		580392-4	NUT	
1	32		580392-1	NUT	
1	31		SK65943-1	SUPPORT, BEARING	
1	30		SK65941-1	SHAFT ASSY, MOTOR	
1	29		SK65939-1	ROTOR ASSY	
1	28		SK65935-1	HOUSING, MOTOR	
1	27		SK65934-1	INSERT, HOUSING	
ONE OF EITHER	26		SK65933-13	SPACER, CARRIER	
	25		-12		
	24		-11		
	23		-10		
	22		-9		
	21		-8		
	20		-7		
	19		-6		
	18		-5		
	17		-4		
	16		-3		
	15		-2		
	14		SK65933-1	SPACER, CARRIER	
ONE OF EITHER	13		SK65931-7	SPACER, SHAFT, PUMP	
	12		-6		
	11		-5		
	10		-4		
	9		-3		
	8		-2		
	7		SK65931-1	SPACER, SHAFT, PUMP	
	6		SK65930-1	COVER, MAGNET	
	5		SK65929-1	CONE, BEARING, FRONT	
	4		SK65928-1	SHAFT ASSY, PUMP	
	3		SK65927-1	HOUSING, BEARING	
	2		SK65926-1	HOUSING, PUMP	
	1		SK65925-1	IMPELLER	
← ASSY				PARTS LIST	

UNLESS OTHERWISE SPECIFIED-BURR CONTROL PER SC553
IDENTIFICATION MARKING PER MC15
STD INTERPRETATIONS PER PIBS

HEAT TREATMENT | PROCESS
HARDNESS AND SPEC | NAME AND SPEC

CONTRACT NO.
DR [signature] 1-8-74
CHK
VALUE ENGR
MATL
STRESS
APPD
APPD
AIRESEARCH APPD
OTHER AGENCY APPD

AIRESEARCH MANUFACTURING COMPANY
A DIVISION OF THE GARRETT CORPORATION
LOS ANGELES, CALIFORNIA

PUMP & MOTOR ASSEMBLY

SIZE D | CODE IDENT NO. 70210 | DWG NO. SK65924
SCALE 1/1 | SHEET | OF

REQD	NEXT ASSY	USED ON
	FINAL	
APPLICATION		

2 CLEAN PER AIR SPEC C38, CLASS I, LEVEL I.
⚠1 MAY BE PURCHASED FROM: THE ADVANCED PRODUCTS CO. 8921 S. SEPULVEDA BLVD LOS ANGELES CALIF.
NOTES:
UNLESS OTHERWISE SPECIFIED

DWG NO. SK65924

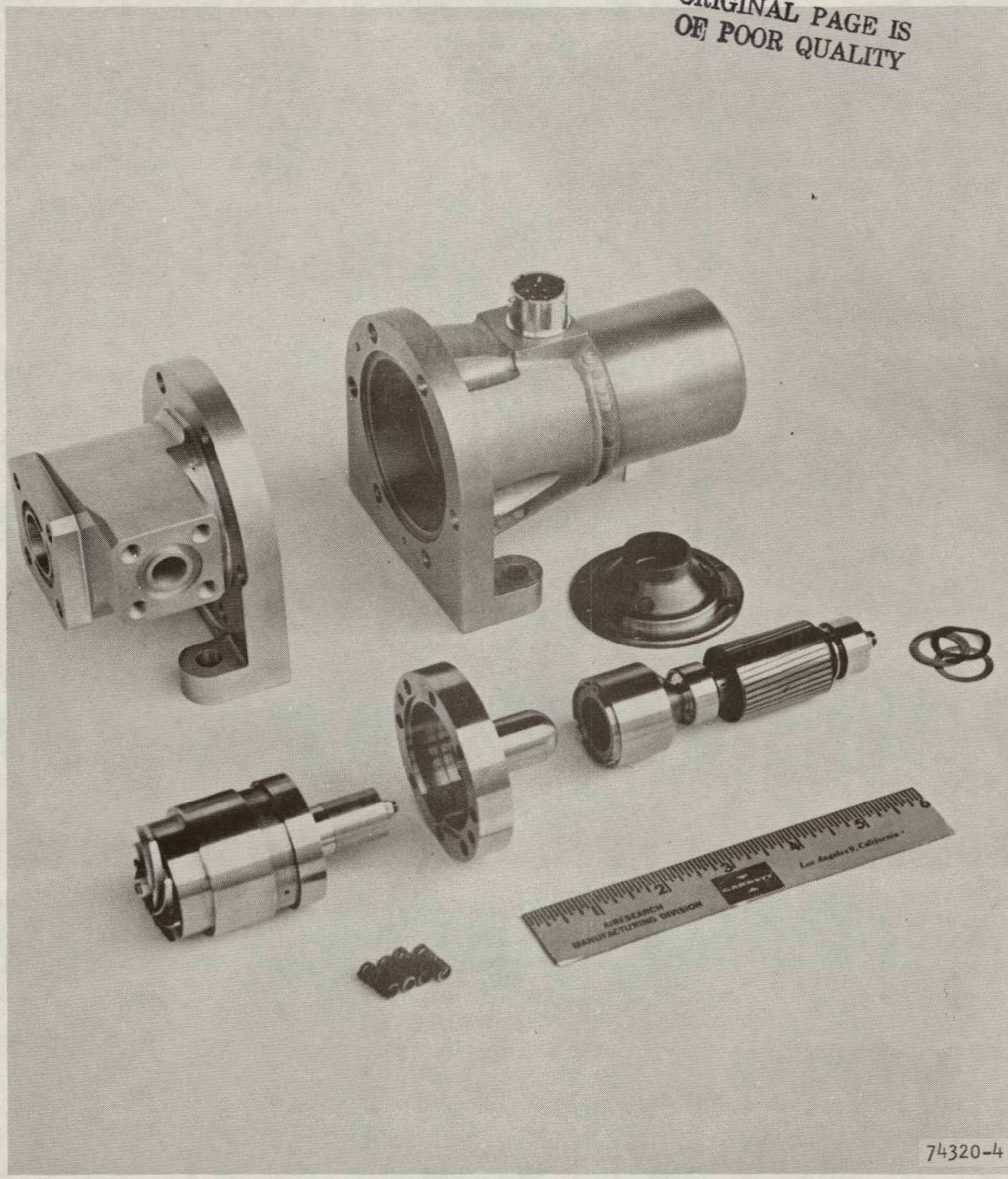

Figure 4-2. Exploded View of Prototype Pump, 581280,
Showing Major Sub-Assemblies.

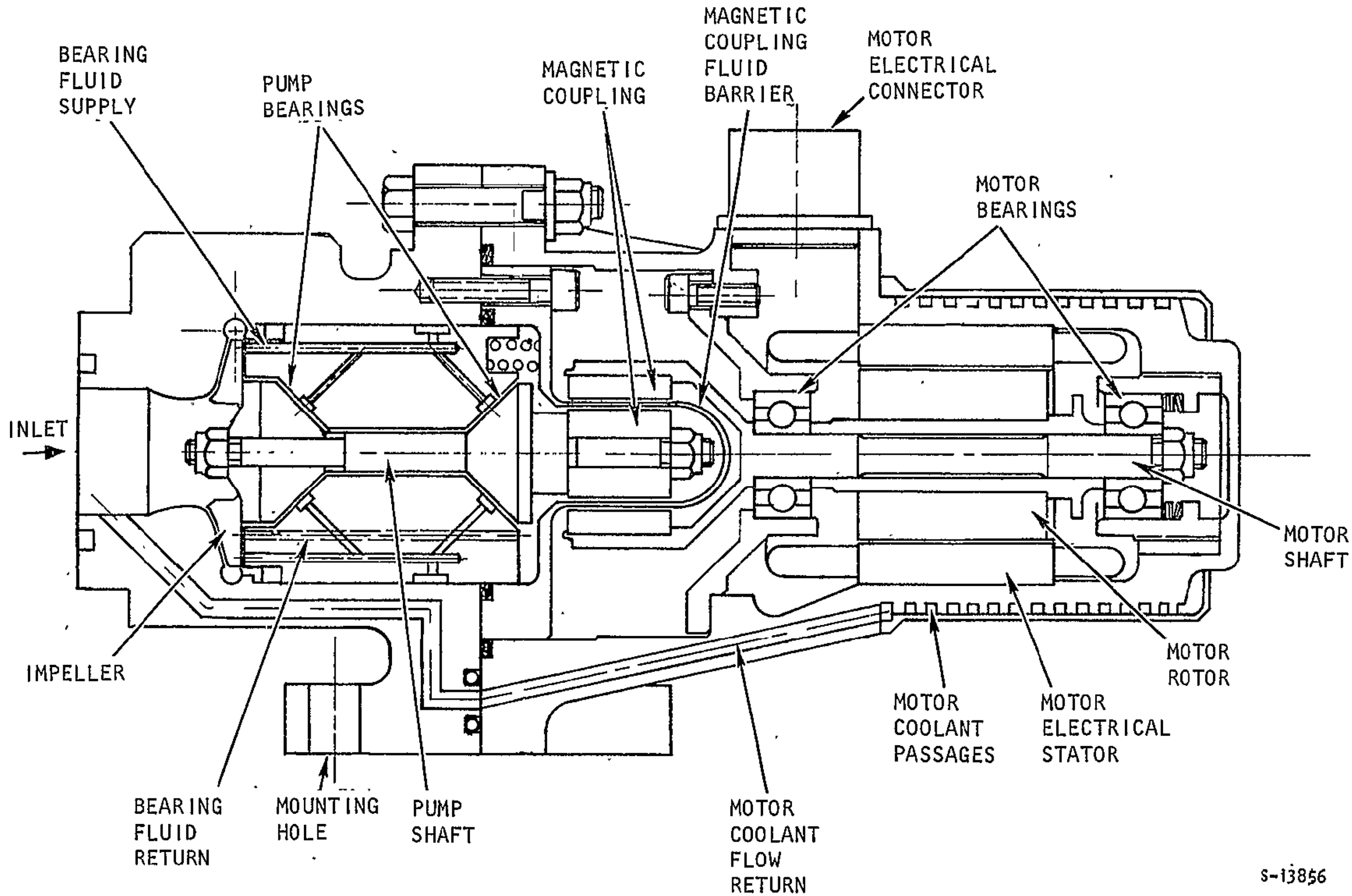

Figure 4-3. Cross Sectional Drawing of Prototype Pump, 581280

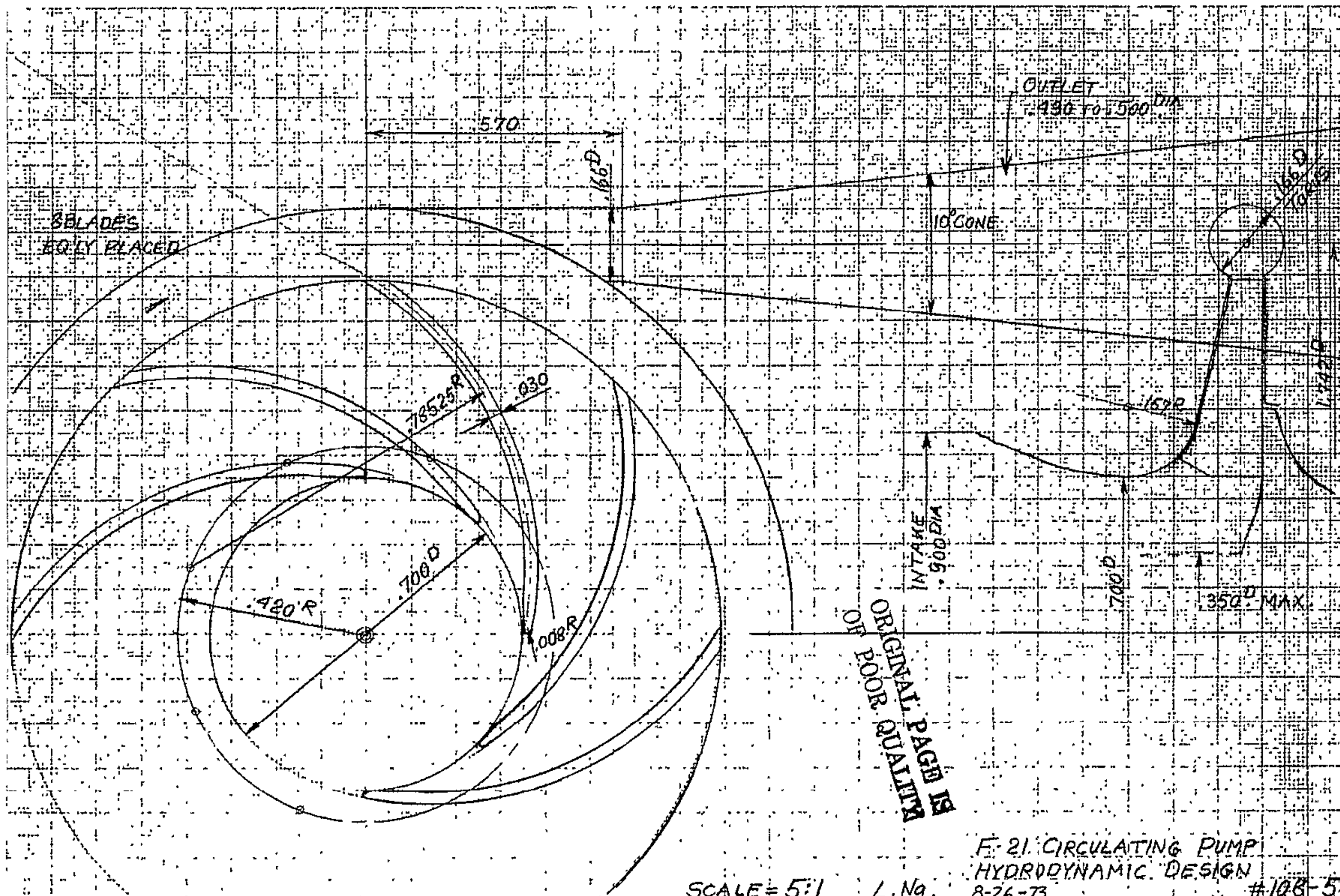

Figure 4-4. Hydrodynamic Design Configuration of Prototype Pump, 581280

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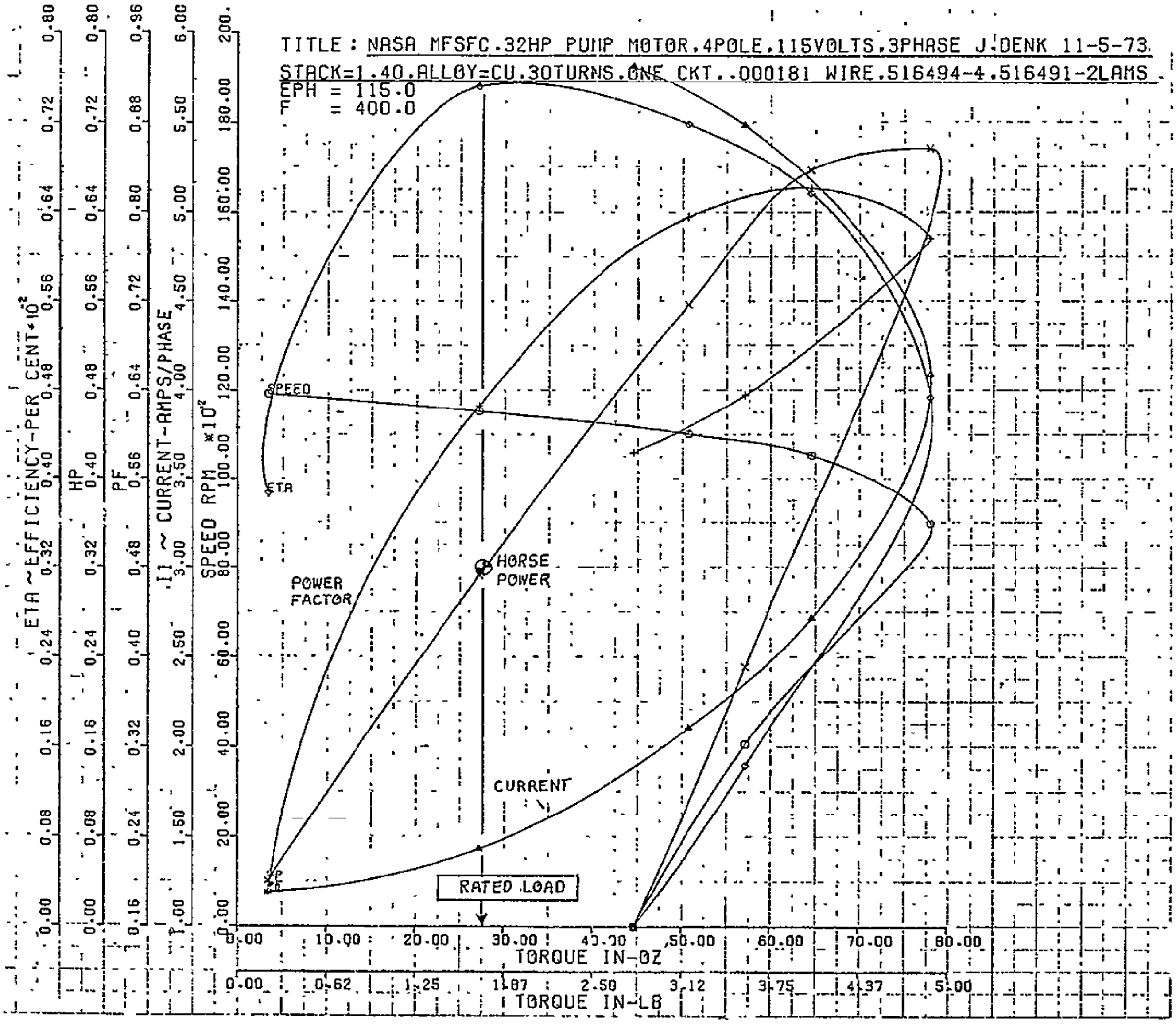

Figure 4-5. Calculated Performance of Motor for Prototype Pump, 581280.

4.2 Test Program

4.2.1 Initial Shakedown Tests

During the initial tests considerable effort was expended to verify the magnetic strength of the inner and outer magnets in the magnetic drive coupling. Eventually a new magnetizing fixture had to be fabricated to obtain the proper magnetic field strength, and this provided satisfactory results.

The conical bearing system was assembled with 0.0017 inches total axial clearance, and pump operation was initially verified using water as the working fluid. Following initial runs with water, the pump was satisfactorily checked out using Freon 21 as the working fluid. During these tests it was determined that the special cooling of the motor stator would not be required, and the coolant passages were blocked off for further work. Also, at this time a speed pick-up was added to the motor shaft. Figure 4-6 shows the prototype pump, 581280, in the Freon 21 test loop.

4.2.2 Calibration Tests

A component calibration test was performed on the motor of the prototype pump, 581280, and the test results are shown in Figure 4-7. A calibration of the complete pump was then performed, and the results of the calibration are shown plotted in Figure 4-8. The resulting hydrodynamic performance was somewhat less than the design target, but was considered acceptable for the bearing development program. The performance can readily be improved for any subsequent requirements, if desired.

4.2.3 Endurance Test

After calibration, the pump was placed on long term endurance test, to determine whether the performance and operation of the pump bearings would change significantly with time. As with the modified ATM pump, 580745, this was essentially an unattended test with pump hydrodynamic and electrical parameters logged once a day until termination of the program. A total endurance time of 10,382 hours and over 130 starts and stops were accumulated during this running, with no significant change in observed operating parameters, except for bearing flow.

4.2.4 Post-Endurance Calibration Test

After termination of the endurance running, a post endurance calibration test was performed, and results of the calibration are shown plotted in Figure 4-9. Comparison of Figure 4-9 with Figure 4-8 shows very little change in pump performance as a result of the endurance test.

4.2.5 Disassembly Inspection

After the post-endurance test, the pump was disassembled for inspection, with findings as outlined below.

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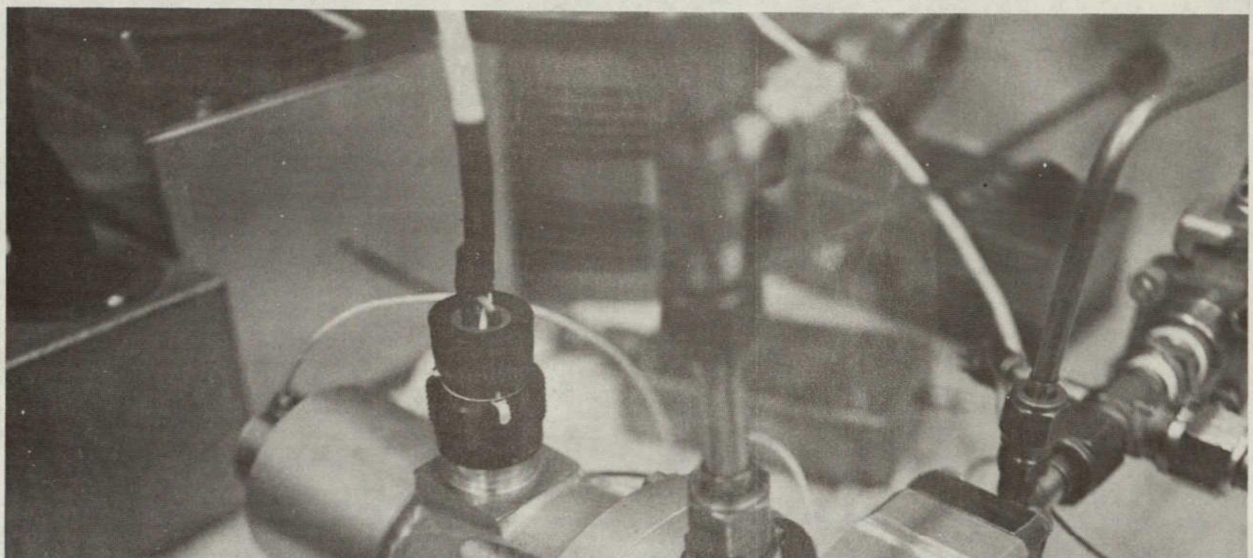

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Figure 4-6. Prototype Pump, 581280, in Freon 21 Test Loop

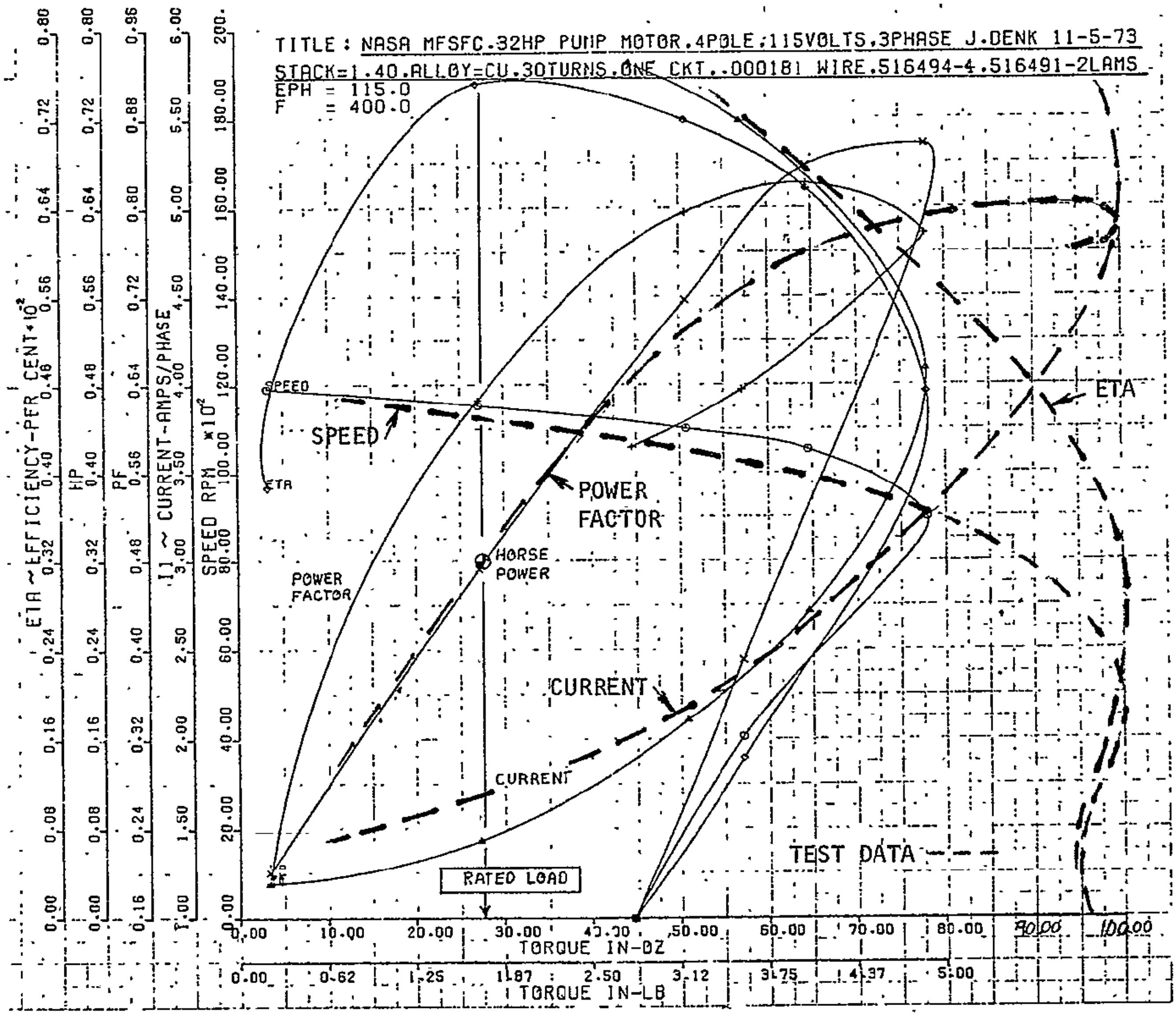

Figure 4-7. Measured Performance of Motor For Prototype Pump, 581280.

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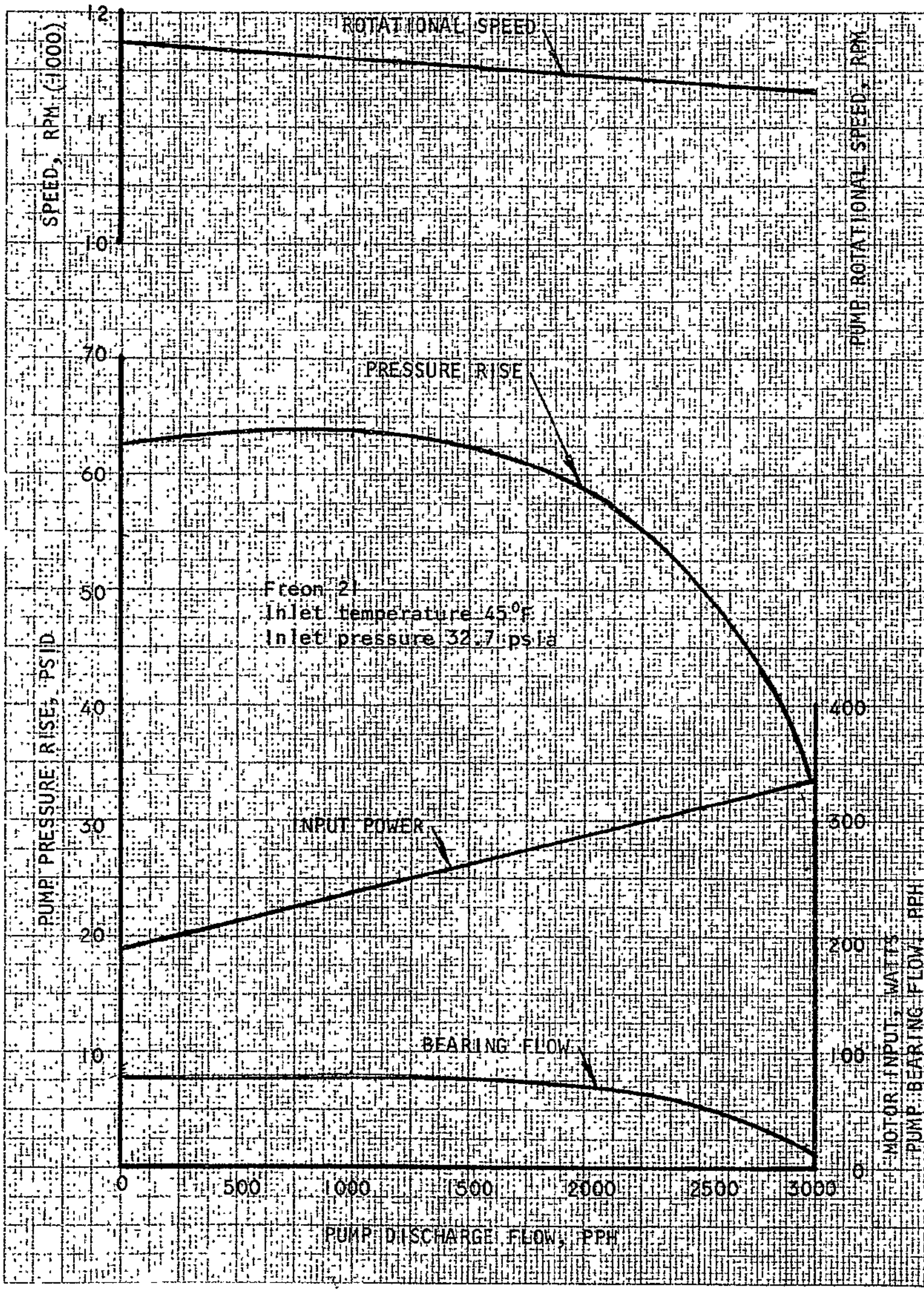

Figure 4-8. Performance of Prototype Pump, 581280, Early in Test Program.

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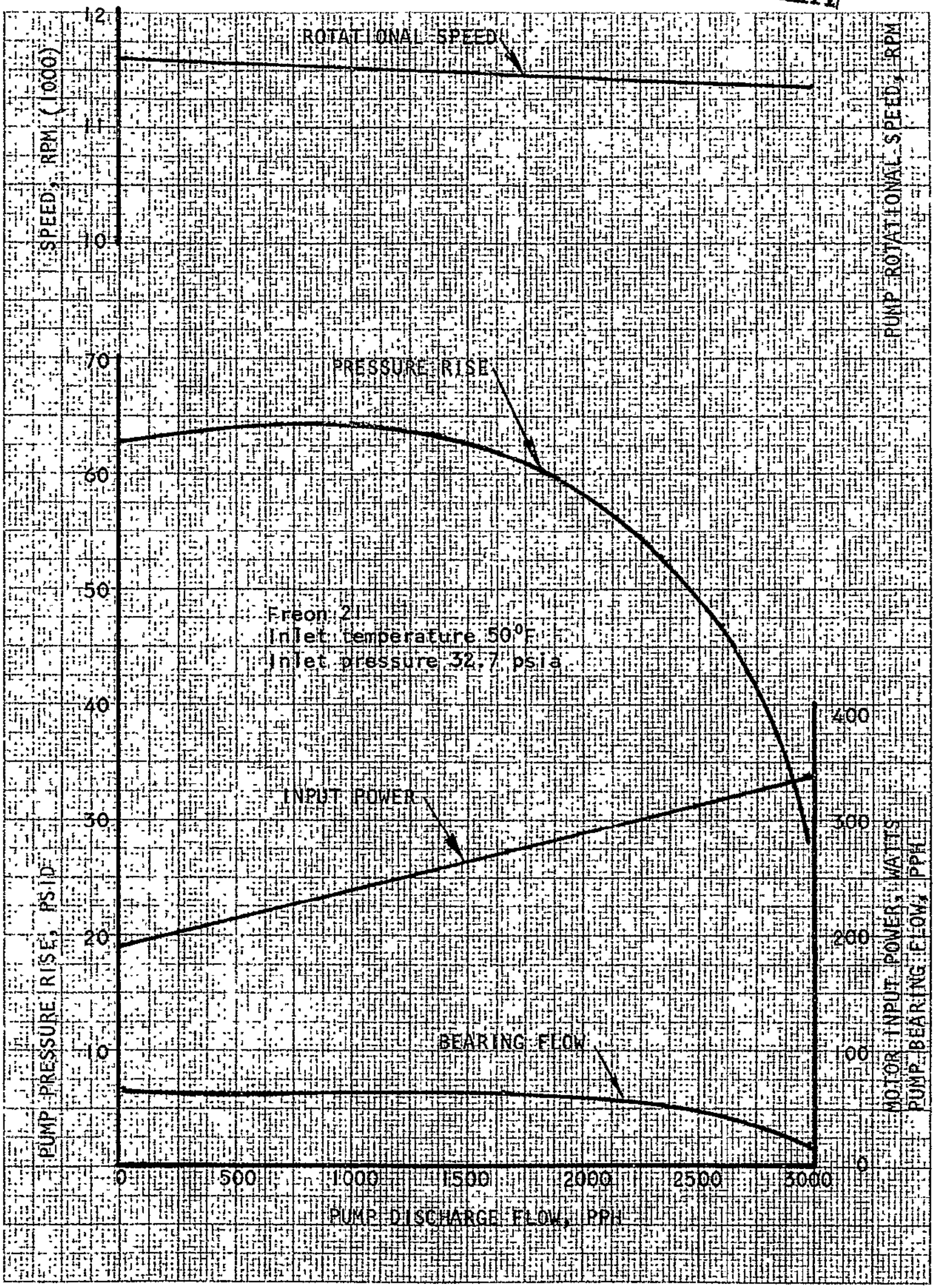


Figure 4-9. Performance of Prototype Pump, 581280, at Completion of Test Program

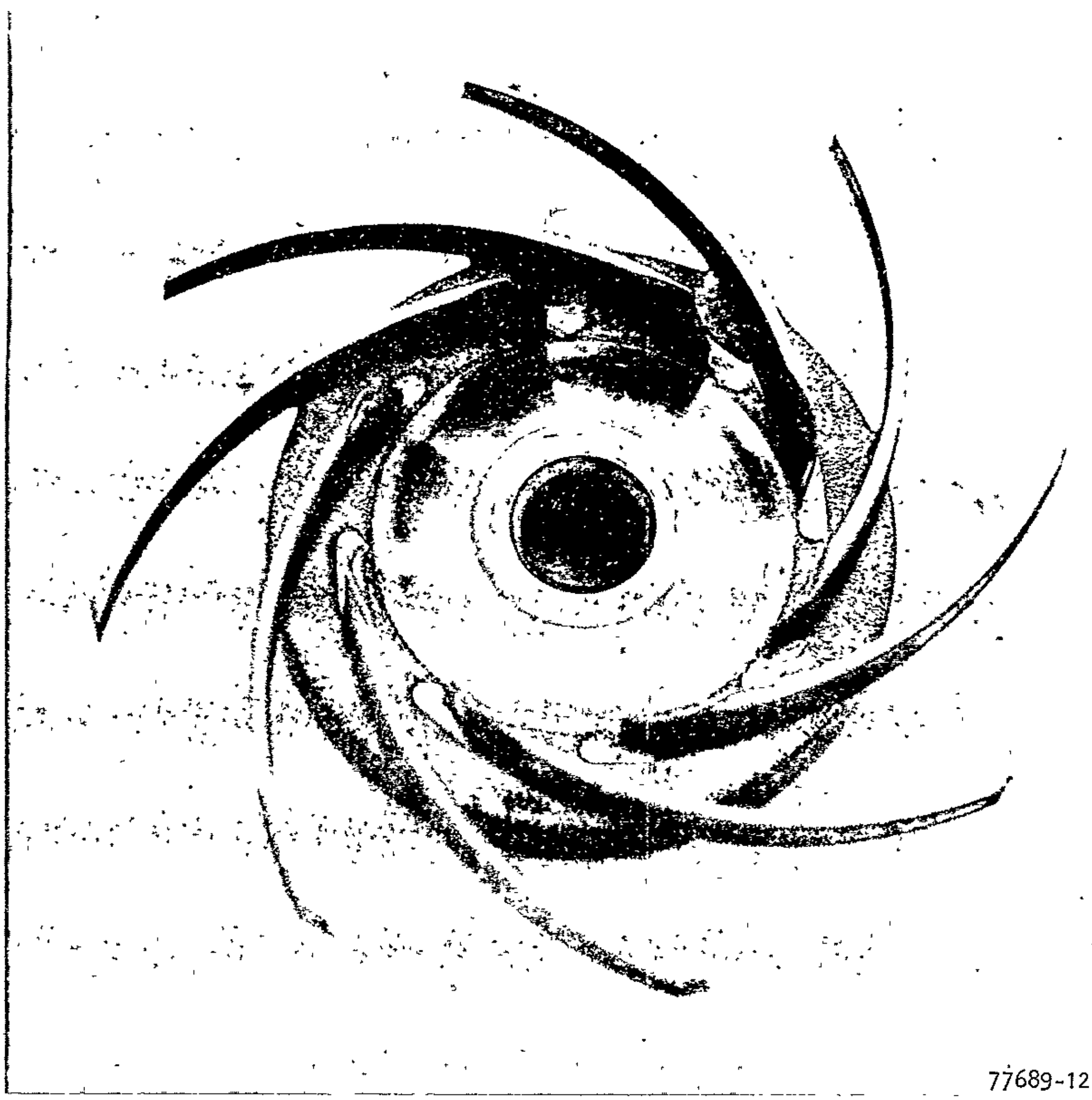
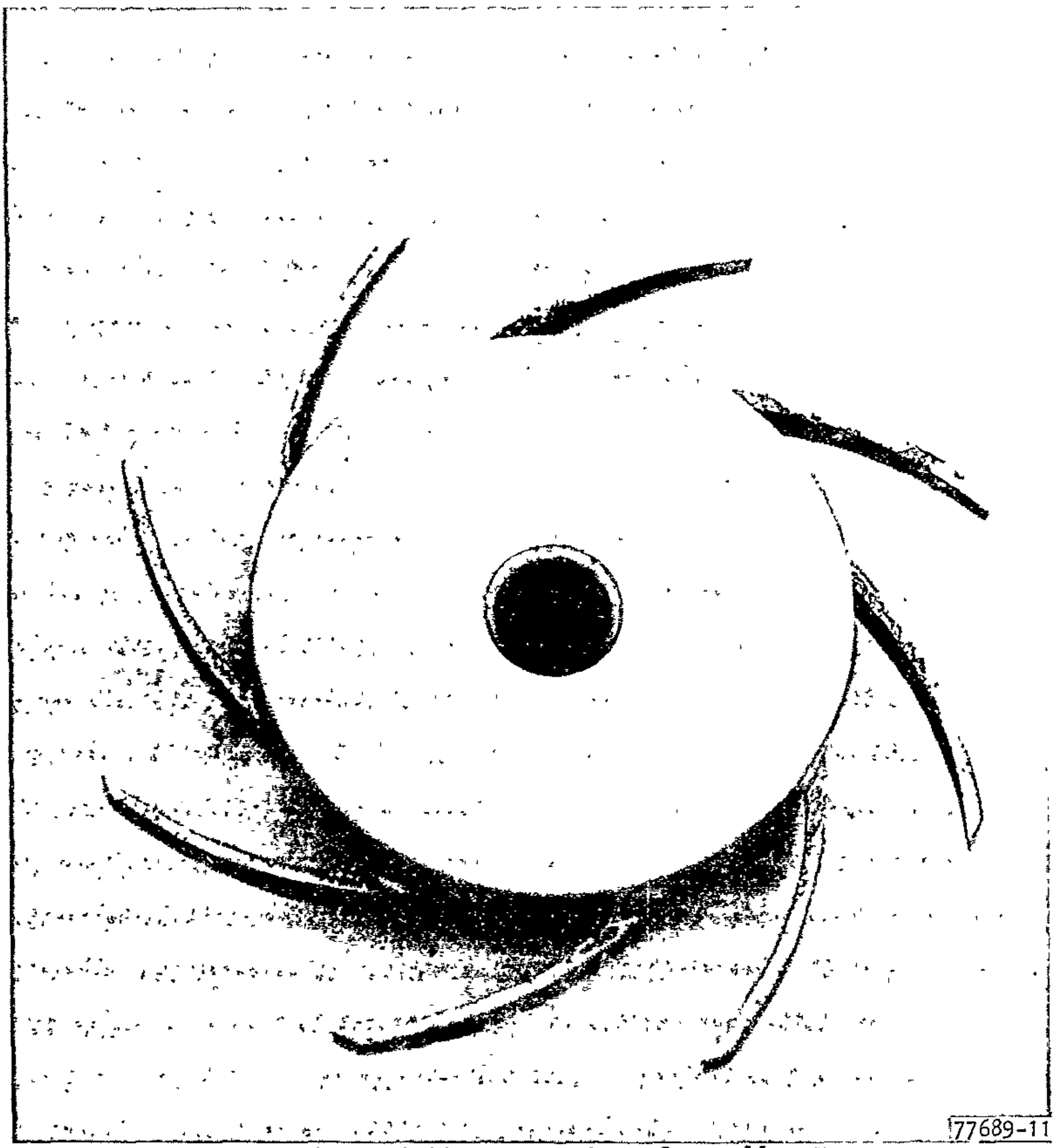


Figure 4-10. Front View of Impeller From Prototype Pump, 581280, After 10,382 Endurance Hours



77689-11

Figure 4-11. Rear View of Impeller From Prototype Pump, 581280, After 10,382 Endurance Hours

Figure 4-10 shows the front of the impeller and Figure 4-11 shows the rear of the impeller after the endurance test. Some light rub marks are shown on the edges of the impeller vanes, and these were probably associated with the rather close face clearances used. A more significant finding is the evidence of some rubbing at the impeller vane tips, probably due to vane deflection, and indicating that the impeller vane thickness should be increased slightly in future efforts.

Figure 4-12 shows the impeller end bearing cone. This shows some rather heavy circumferential contact marks. It is believed that these occurred during starts and stops, when the pressurant flow was low. A profilometer measurement showed the finish to be RMS 45, compared to the initial value of RMS 8.

Figure 4-13 shows the impeller end bearing cone. This also has some rather heavy circumferential contact marks, believed to be associated with the starts and stops. A profilometer measurement showed the finish to be RMS 24, compared to the initial value of RMS 8.

Figure 4-14 shows the impeller end female bearing cone. Some circumferential contact marks are evident, and this was primarily a wearing away of the teflon coating. A profilometer measurement showed the finish to be an average of RMS 20 on the average, except for the several grooves, and this compared to an initial value of RMS 8.

Figure 4-15 shows the magnet end female bearing cone. Some circumferential contact marks are evident, and this is primarily a wearing away of the teflon coating. A profilometer measurement showed the finish to be an average of RMS 20 except for the several grooves, and this compared to the initial RMS 8.

Figure 4-16 shows the motor bearings after test. The grease shield of the bearing in the right of the photograph shows a deposit of varnish from the electrical stator winding. This resulted from a Freon 21 leak into the motor cavity caused by a leak in the gasket used to block off the motor coolant passages for test purposes. The bearings were in excellent condition as described by the bearing disassembly inspection report shown in Figure 4-17. The bearings showed very little change after 10,382 hours, and it is estimated that the bearings could operate another 10,000 hours or more, judging by the excellent condition of the grease.

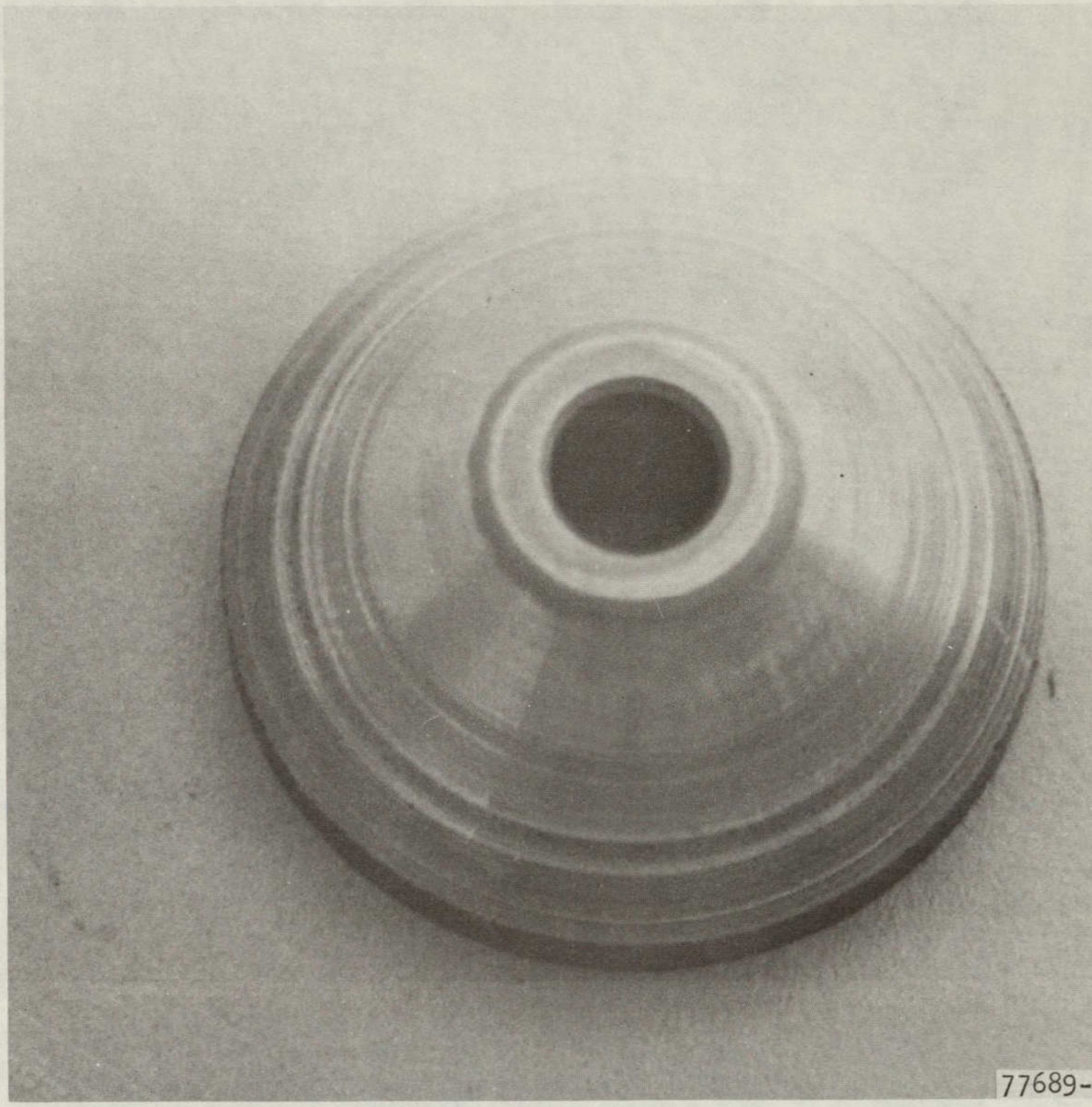


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Figure 4-12. Impeller End Bearing Cone
From Prototype Pump, 581280,
After 10,382 Endurance Hours.

Figure 4-13. Magnet End Bearing Cone
From Prototype Pump, 581280,
After 10,382 Endurance Hours.

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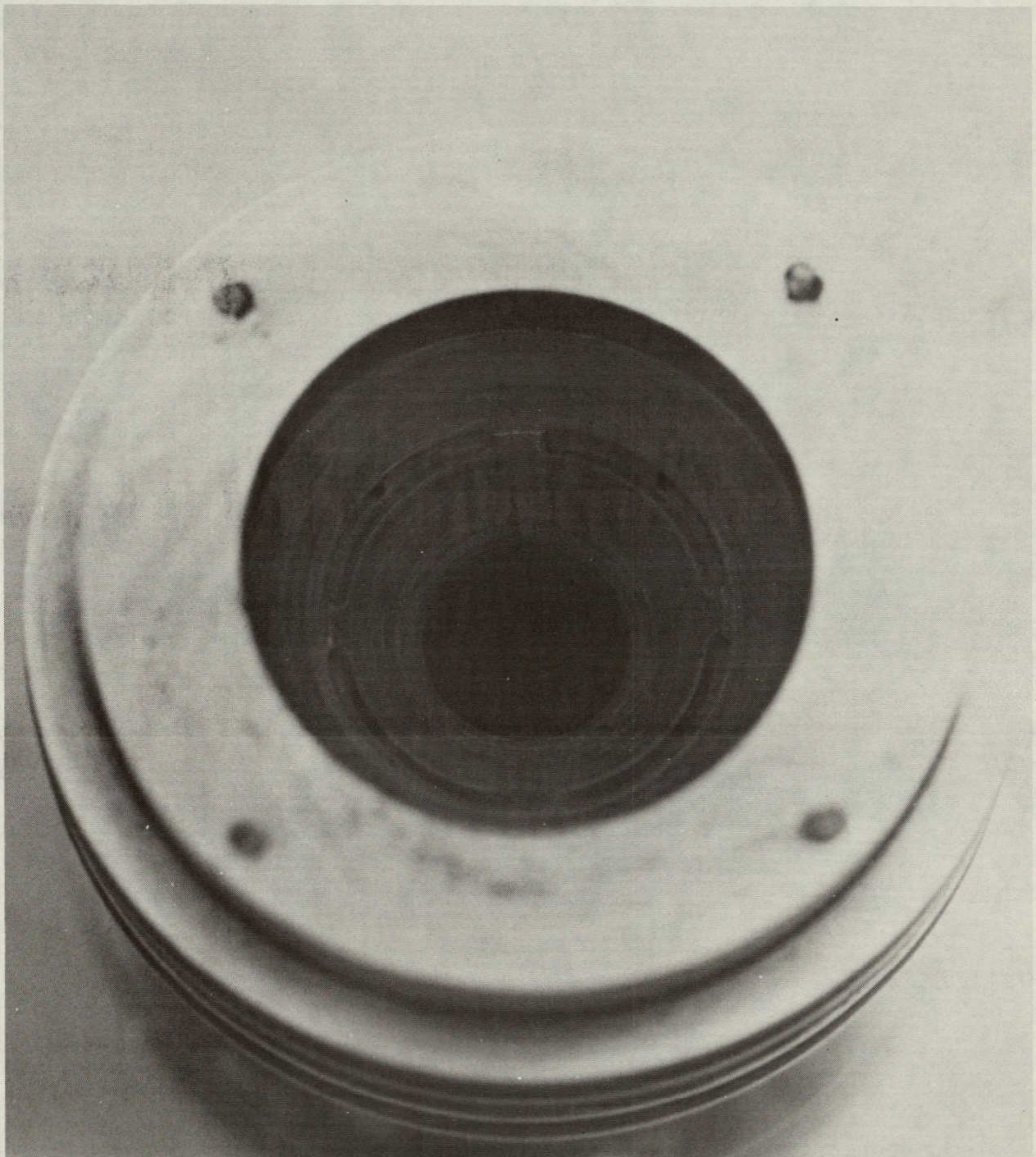
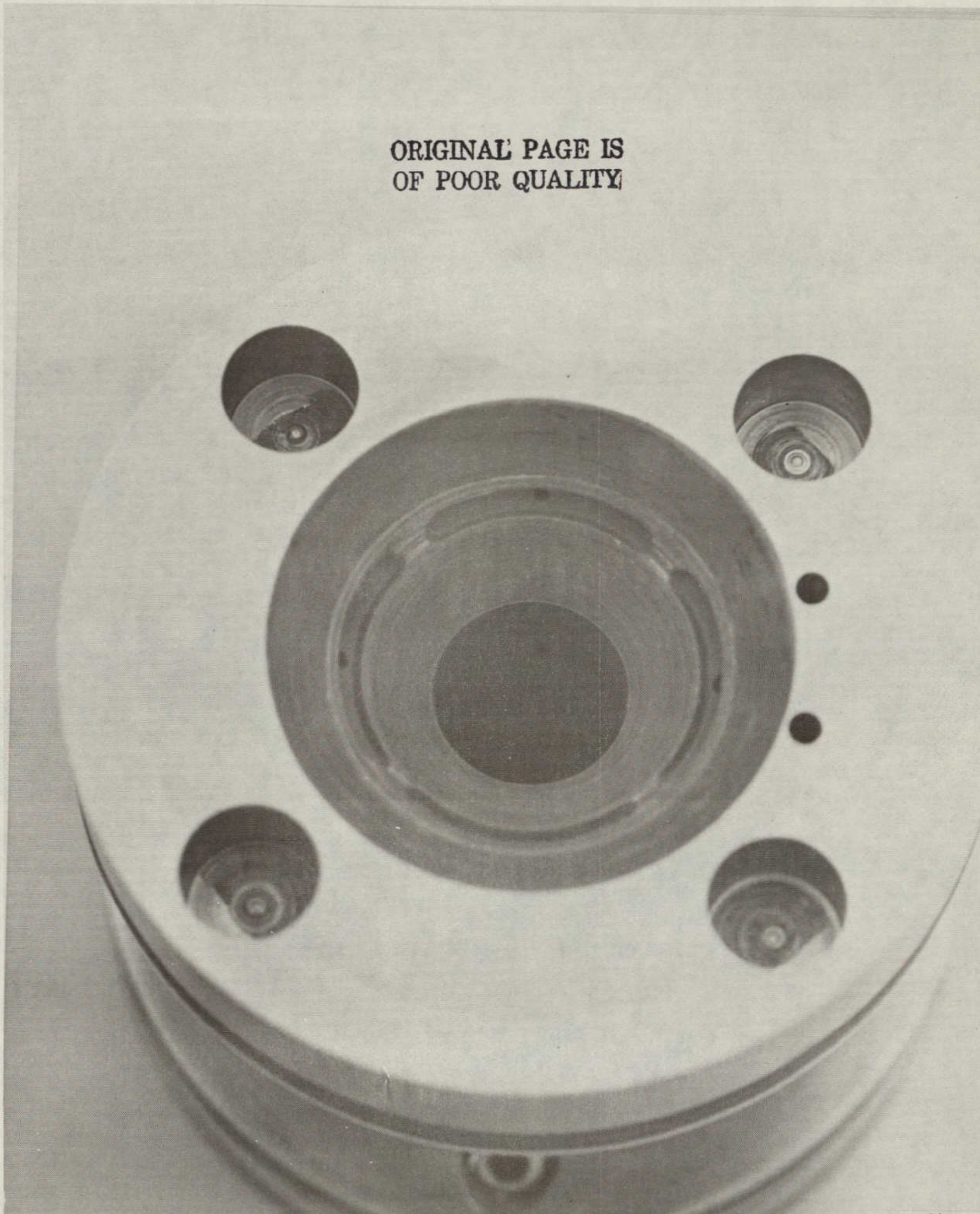


77689-13

Figure 4-14. Impeller End Female Bearing Cone From Prototype Pump, 581280, After 10,382 Endurance Hours.



77689-16

Figure 4-15. Magnet End Female Bearing Cone
From Prototype Pump, 581280,
After 10,382 Endurance Hours.

Figure 4-16. Motor Bearings, 580629-1, After 10,382 Hours of Operation in Prototype Pump, 581280.

OPERATING DATA

10382 HOURS

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BEARING REPORT BR770308-8-1623

UNIT

P/N SK 65924

S/N

BEARING BARDEN

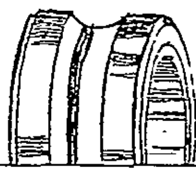
DIST. J. RIPLE

R.O. 300-08-77-8864

NO. I MAGNET END BEARING S/N 22 .171 GRAM GREASE IN THE BEARING

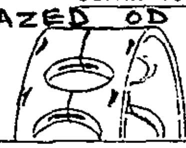
INNER RING

BRIGHT SURFACES. FAINT LIGHT LOAD TRACK.



SEPARATOR

GLAZED OD AND POCKET SURFACE.

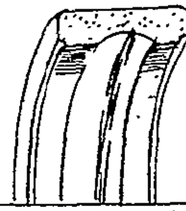


BALLS

BRIGHT SURFACES.

OUTER RING

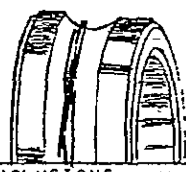
OD CREPT IN THE HOUSING. BRIGHT SURFACES. FAINT LIGHT LOAD TRACK.



NO. II REAR END BEARING S/N 25 .128 GRAM GREASE IN THE BEARING.

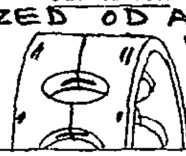
INNER RING

BRIGHT SURFACES. FAINT LIGHT LOAD TRACK.



SEPARATOR

GLAZED OD AND POCKET SURFACE.

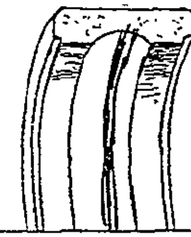


BALLS

BRIGHT SURFACES.

OUTER RING

MEDIUM FRETTING OF OD AND OUTBOARD FACE. BRIGHT SURFACE. FAINT LIGHT LOAD TRACK.



CONCLUSIONS: THE BEARINGS ARE IN EXCELLENT CONDITION. THE RACEWAY, BALL AND SEPARATOR SURFACES SHOW NO WEAR OR HEAVY LOADING OR ADVERSE OPERATING CONDITIONS.

THE GREASE IN THE BEARINGS IS EVENLY DISTRIBUTED, SMEARS READILY AND SHOWS NO DEGRADATION.

THE BEARINGS APPEAR TO BE AS GOOD AS NEW AND WOULD PROBABLY OPERATE FOR 10000 MORE HOURS OR MORE.

CONDITION	I	II
BALANCE		
LOADS		
ALIGNMENT		
LUBRICATION		

OD AND FACE FRETTING OF REAR BEARING IS DUE TO VIBRATION OR LOOSE HOUSING FIT.

3-8-77
DATE

R. Bhikha
BEARING COORDINATOR

Figure 4-17. Results of Inspection of Bearings from Prototype Pump, SK 65924, after 10,382 Hours of Operation

5. DISCUSSION OF RESULTS

The hydrodynamic analysis performed in the design concepts study showed that a significant improvement in hydrodynamic efficiency, increasing from 0.53 to 0.69, can be achieved by using a shrouded impeller with 3 dimensional blades, although the manufacturing complexity and cost would be greater. In addition, it is probable that further improvements in efficiency can be achieved by operating at a somewhat higher specific speed and rotational speed. Since this would result in a significant reduction in impeller size, much tighter control of geometric tolerances would be required to avoid the usual significant scale efficiency penalties associated with such a reduction in size. While these potential improvements in efficiency would result in increased manufacturing cost, they may be warranted for power limited space vehicle missions.

In the investigation, it was assumed that pump inlet pressure could be provided as required by the vehicle system and, accordingly, particular attention was not paid to suction performance design. For systems having a constraint on pump inlet pressure, some improvement in pump suction performance can be obtained by the addition of an inducer to the pump impeller and/or jet pumping the inlet from pump discharge. This latter has a power penalty associated with it.

Regarding changes in pump head or flow to fit particular applications, within reasonable limits these can be obtained by tailoring the pump geometry to the new requirements.

The all metal pressurized, double conical bearing was shown to be feasible for long life coolant pumps, and a decided advantage for use with fluids having a tendency to swell or otherwise dimensionally distort non-metallic materials. Some improvement in bearing materials and coatings to aid operation during the boundary lubrication condition which occurs during stops and starts is desirable. In addition, some further investigation of bearing geometry appears warranted.

The basic design arrangement of dry motor, grease packed ball bearings, magnetic coupling, and centrifugal pump was shown to be a sound design arrangement for the space vehicle coolant pump application.

The impeller vanes of the prototype pump, 581280, appeared to be deflecting to an undesirable degree during operation, and a slight increase in blade thickness is desired if the impeller is unshrouded.

The grease packed bearings in the motors were shown to be capable of operating 2 to 3 times as long as they were operated during the endurance tests, i.e., capable of 20,000 to 30,000 hours.

Based upon the results of the tear down inspection, it was estimated that either pump, as built, could have operated satisfactorily for at least a two year duration, i.e., 17,520 hours. With some design improvements, they should be capable of longer operation.

6. CONCLUSIONS

Conclusions made at the end of the program were:

- The all metal pressurized double conical bearing was shown to be feasible for long life coolant pumps, and a decided advantage for use with fluids having a tendency to swell or otherwise dimensionally distort non-metallic materials.

- Some improvement in pump hydrodynamic efficiency is achievable with configurations having greater manufacturing complexity. Such configurations may be warranted for power limited space applications.

- The magnetic coupling approach continues to demonstrate its viability as a means of eliminating shaft dynamic seals, of isolating the motor elements from fluids which may be active solvents such as Freon 21, and of minimizing rotating element fluid viscous losses, compared to "canned" motor approaches.

- Grease packed bearings for the motor electrical rotor, with a properly designed installation, show very little change after 12,000 hours operation.

- Based upon an overall assessment of the results of the program, it is concluded that reliable coolant pumps can be designed for three year space missions (26,280 operating hours) with a high confidence level.

7. RECOMMENDATIONS

Based upon the results of the investigation and upon the current state of several other significant technological developments, the following actions are recommended:

7.1. Flight Version of Prototype Pump

Prepare a flight weight design of the prototype pump, fabricate, and test in the significant areas which were not covered in the program to date. These additional test areas include vibration and acceleration tests, plus any other significant tests required for a particular intended application. Such a program, representing a modest amount of additional effort, would provide a coolant pump design of demonstrated life capability, suitable for long duration space missions.

7.2 Continued Advanced Technology Program

Several recently developed advanced design concepts have the potential for significantly reducing typical space vehicle coolant pump size, weight, and electrical input power, while retaining a long life capability. Such improvements naturally would have a significant impact on the design of space vehicle coolant systems. Accordingly, it is recommended that these concepts be evaluated and demonstrated in a "next generation" prototype design pump, to establish and verify their range of applicability. These recommendations include:

- Investigate the use of a brushless DC motor of advanced design, possibly operating at a higher rotational speed than that representative of current coolant pump design practice. This recommendation is based upon recent advances in brushless DC motor technology, and in solid state electronics, which have produced highly efficient motor systems having excellent reliability.

  The brushless DC motor would be a permanent magnet type, having a basic motor efficiency of 0.91 compared to the 0.65 obtainable with a 3 phase AC motor in the size range typical for space system coolant pumps.

  The efficiency of the front end electronics for the brushless DC motor is approximately 0.82, which is equivalent to the efficiency of the DC to AC inverter required for AC motors operating on the typical space vehicle DC power system. The resulting efficiency product of the electronics plus motor is then 0.75 for the brushless DC system, and 0.53 for the 3 phase AC system. This represents a 42% improvement in the efficiency of the motor and electronics, obtained by using the brushless DC motor.

  The brushless DC motor can operate at a preferred rotational speed determined by the pump hydrodynamics, and is not constrained to certain particular speeds determined by the number of motor poles and the electrical input frequency, as is the AC induction motor. In some cases this can be a distinct advantage if the hydrodynamic requirements place the pump optimum specific speed (best efficiency

design) and desired rotational speed at a speed not readily achieved by the AC induction motor operating at the available frequency. Also, it should be noted that the motor weight and size are approximately inversely proportional to rotational speed, presenting the potential for significant reductions in size and weight, if it should prove feasible to operate at higher design rotational speeds.

In addition, the speed of the brushless DC motor can be varied easily by changing voltage level. This offers the potential for modulating system cooling as required, by varying pump speed, while saving electrical power during those periods when maximum cooling flow is not desired.

Another advantage is that the brushless DC motor does not have a significant amount of heat generated in the rotor, as AC motors have. This greatly simplifies the problem of keeping the motor bearings cool, which is one of the first requirements for long life of the motor grease packed bearings.

- Investigate designing the pump hydrodynamics for a somewhat higher specific speed. A specific speed of approximately 1500 is recommended, and this shows a higher theoretical efficiency. However, it results in a smaller diameter, higher speed, impeller and the "normal" estimate of the reduction in efficiency due to the size scale effect predicts that the higher efficiency will not be achieved with normal manufacturing techniques. The use of improved manufacturing techniques and much closer control of the geometric tolerances of the pump hydrodynamic elements offers the potential of significantly reducing the magnitude of the size scale effect, permitting the actual achievement of a higher efficiency coolant pump of smaller size. The higher manufacturing cost of such a pump may be warranted by power limited space vehicle applications.

- Investigate the use of an advanced samarium cobalt alloy for the magnetic coupling, instead of platinum cobalt. The samarium cobalt alloy has a magnetic strength maximum energy product (BH_{max}) of 23×10^6 Gauss-Oersteds, compared to 9.5×10^6 Gauss-Oersteds for the platinum cobalt alloy. This permits reducing the size parameter (D^2L) of the magnet system in direct proportion to the increase in energy product, i.e., a reduction of 59%. Such a reduction in magnet dimensions significantly reduces the viscous loss associated with the pump magnet rotating in the working fluid, and it is expected that a worthwhile increase in pump overall efficiency can be achieved.

  The samarium cobalt alloy is more active chemically than the platinum cobalt, so a protective coating or plating will be required. A cost advantage is the fact that the samarium cobalt alloy is less expensive than the platinum cobalt by a factor of 15, and this can result in a very significant reduction in pump cost.

- Investigate further design improvements to the pressurized bearing used for the pump rotating group. In particular, the pressurized journal bearing and flat configuration thrust bearing should be evaluated in comparison with the double conical configuration, since it may significantly reduce pump break-away torque, thus improving the starting operational margin. In addition, it has the potential for reducing the sensitivity of pump bearing operation to pump bearing system axial clearance, and it would simplify manufacture.

Properly worked out and integrated into a pump design, these advanced design concepts offer the potential for achieving a superior "next generation" coolant pump for long duration space missions.

APPENDIX A

MOTOR THERMAL ANALYSIS METHOD

PROTOTYPE PUMP, 581280

A detailed thermal analysis has been performed on the pump-motor assembly. The temperature distribution was calculated using the thermal analyzer program (H0298). The internal heat generation due to windage friction in the motor and the journal bearing was calculated by the program. The ball bearing heat generation was calculated using the bearing heat generation program. The electrical losses (I^2R) as a function of conductor temperature (or resistance) was considered in the motor analysis.

The conditions used in the analysis are summarized below.

- Motor input = 310 watts at 11,500 rpm rated load
- Pump flow = 3.8 gpm Freon-21
- Coolant flow to motor = 0.076 gpm (2% pump flow)
- Flow to journal bearing = 0.076 gpm (2% pump flow)
- Flow temperature = 120°F

It was assumed that no radiation and convection heat exchange from the assembly to the surroundings. The geometry of the heat exchanger around the motor housing is calculated based on the preliminary pressure drop requirement (20 psi) and 2% (0.076 gpm) of coolant flow. The required heat exchanger is a 16-turn double lead spiral (approx. 2.1 in. OD) groove with 0.07 in. wide and 0.06 in. deep. The inlet and outlet of the heat exchanger can be located at the pump end of the motor with the double lead spiral groove.

The temperatures and heat generation at the critical locations of the assembly are summarized in Table 1. Figures 1 and 2 present the thermal nodal network of the pump-motor used in the analysis. A complete all nodal temperatures are included in Table 2.

TABLE I

NASA MSFC PUMP-MOTOR ASSEMBLY TEMPERATURE SUMMARY

- Total pump flow = 3.8 gpm Freon-21
- Coolant flow to motor = 0.076 gpm (2% pump flow)
- Fluid temperature = $120^{\circ}F$
- ΔP through heat exchanger = 20 psi
- Motor input = 310 watts at 11,500 rpm rated load

Locations	Node No.	Heat Generation, watts	Temperatures	
			$^{\circ}F$	$^{\circ}C$
Motor				
Stator end turns	1,2	28.0	198	92
Stator stack winding	3	22.8	184	85
Stator tooth	4	2.2	155	68
Stator back iron	5	2.6	145	63
Rotor bar	8	8.0	197	92
Rotor end rings	6,7	3.0	198	93
Rotor tooth	9	0.2	197	92
Rotor iron	10	0.4	197	92
Left bearing	50	2.4	147	64
Right bearing	51	2.4	150	66
Air gap	56	0.1	184	85
Heat exchanger	22-24	0	136	58
Exit fluid	85	0	135	57
Pump				
Pump blades	11	0	120	49
Journal bearing	30	0	127	53
Fluid gap	74		124	51
Fluid gap	76	4.7	123	50
Fluid gap	77		125	52
Magnet	48	0	141	60
Magnet	49	0	125	52
Shield	39	0	125	52

Total losses = 76.8 watts

Figure 1. NASA-MSFC Pump-Motor Assembly (Numbers designate nodal element locations for thermal analysis) Approx. full size.

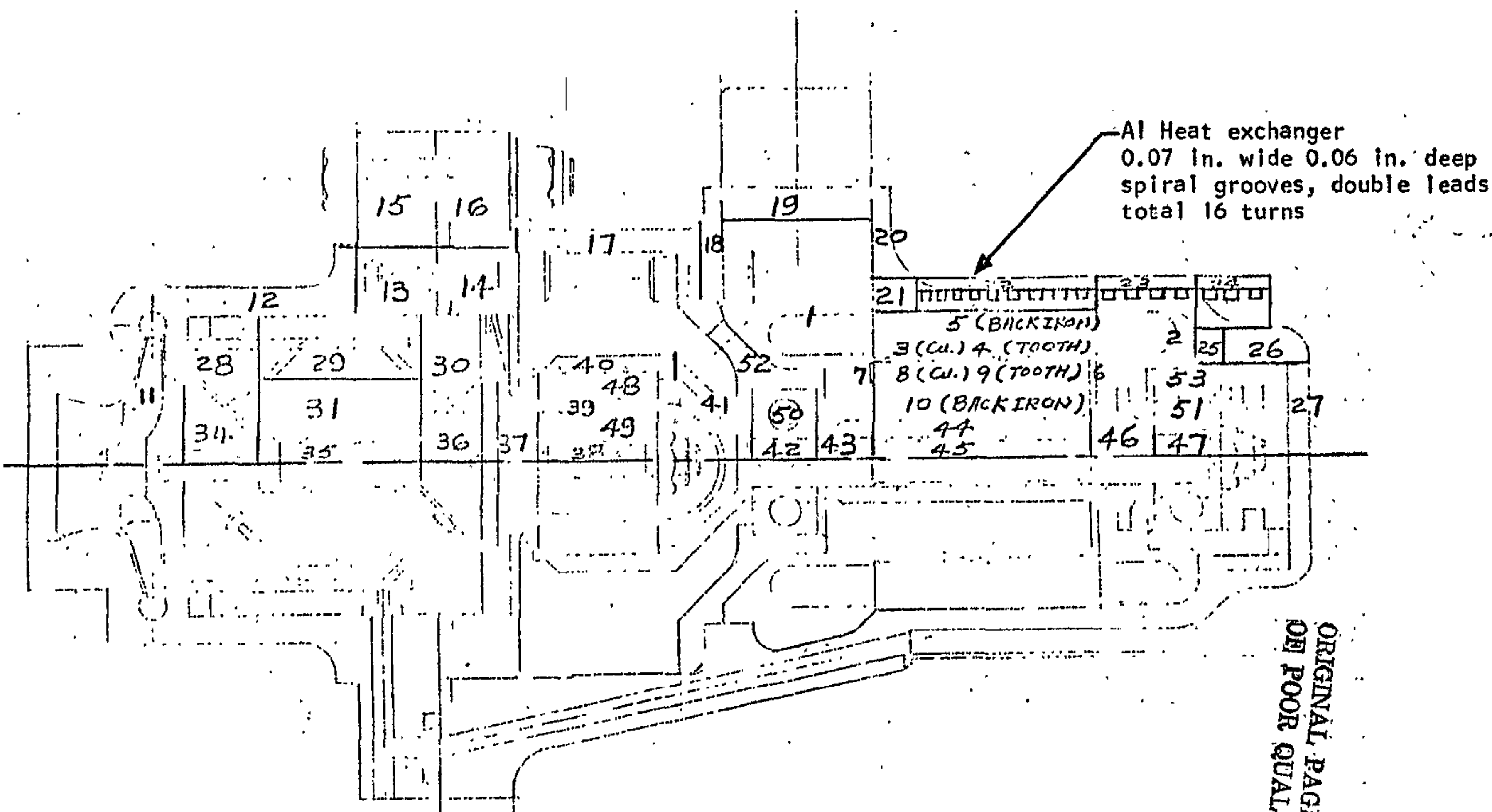


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Figure 2. NASA MSFC Pump Motor Assembly (Numbers designate fluid stream elements for thermal analysis)

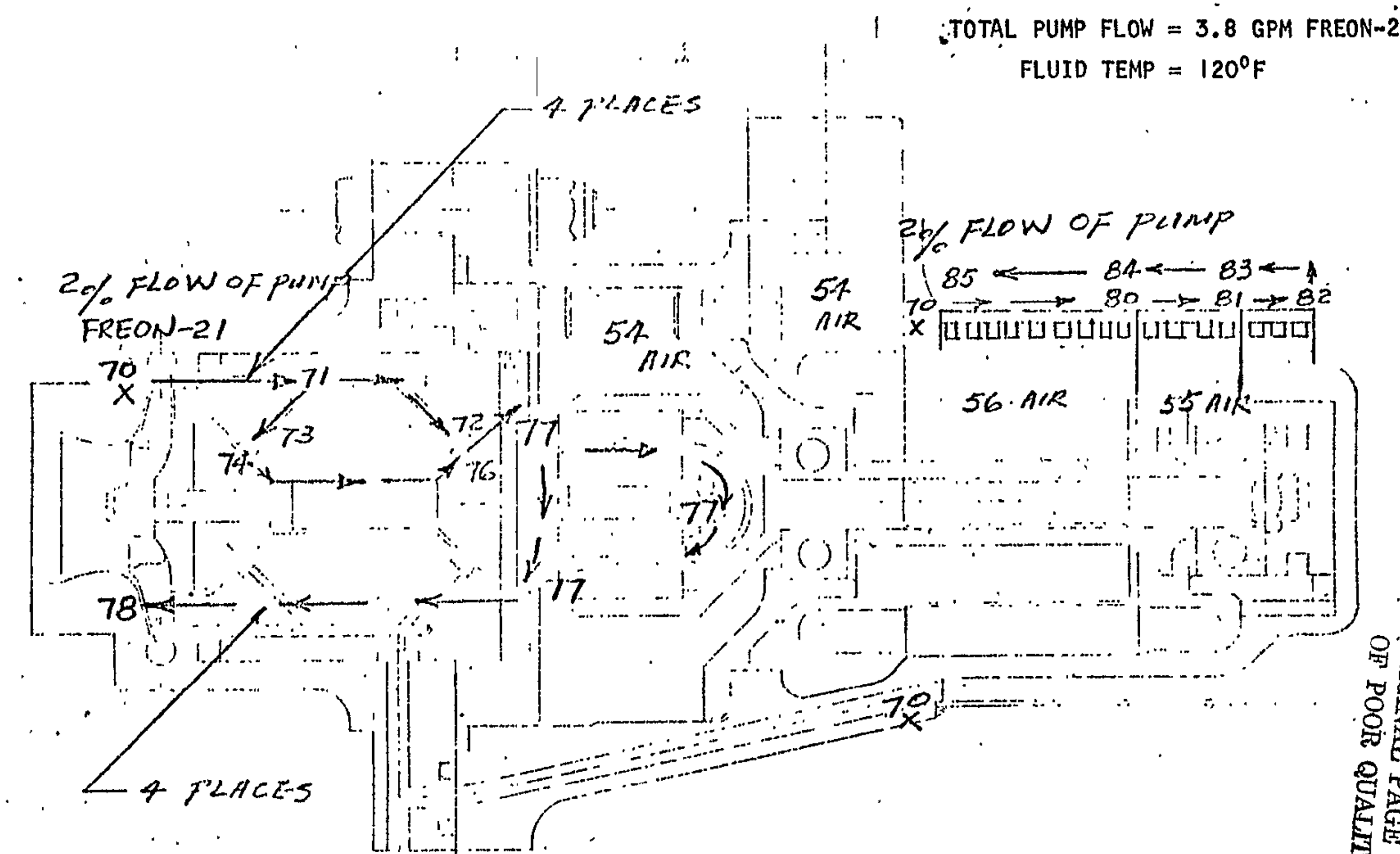

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TABLE 2

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STEADY STATE SOLUTION
NO.OF ITER.= 100 DTEMP= .5000 ACCEL= .2059

NODE NO.	TEMP.F	WATTS HEAT IN.	RHOV	CPN	KN	TEMPC.
1	197.28	14.3725	.000000	.0000	200.00000	91.81
2	198.58	14.4049	.000000	.0000	200.00000	92.53
3	184.22	22.7948	.000000	.0000	200.00000	84.56
4	154.50	2.2000	.000000	.0000	12.00000	68.04
5	145.27	2.6000	.000000	.0000	12.00000	62.91
6	197.75	1.4941	.000000	.0000	200.00000	92.07
7	197.43	1.4933	.000000	.0000	200.00000	91.89
8	197.01	7.9589	.000000	.0000	200.00000	91.66
9	196.97	.2000	.000000	.0000	12.00000	91.64
10	196.66	.4000	.000000	.0000	12.00000	91.46
11	120.00	.0000	-.000579	1.0000	1.00000	48.88
12	128.77	.0000	.000000	.0000	100.00000	53.75
13	129.23	.0000	.000000	.0000	100.00000	54.01
14	129.39	.0000	.000000	.0000	100.00000	54.09
15	129.52	.0000	.000000	.0000	100.00000	54.16
16	129.65	.0000	.000000	.0000	100.00000	54.24
17	131.67	.0000	.000000	.0000	100.00000	55.36
18	133.22	.0000	.000000	.0000	100.00000	56.22
19	134.03	.0000	.000000	.0000	100.00000	56.67
20	135.83	.0000	.000000	.0000	100.00000	57.67
21	136.42	.0000	.000000	.0000	100.00000	58.00
22	136.04	.0000	.000000	.0000	100.00000	57.79
23	135.03	.0000	.000000	.0000	100.00000	57.23
24	135.14	.0000	.000000	.0000	100.00000	57.29
25	135.32	.0000	.000000	.0000	100.00000	57.39
26	135.31	.0000	.000000	.0000	100.00000	57.38
27	135.60	.0000	.000000	.0000	100.00000	57.54
28	126.27	.0000	.000000	.0000	10.00000	52.36
29	127.44	.0000	.000000	.0000	10.00000	53.01
30	126.78	.0000	.000000	.0000	10.00000	52.64
31	125.93	.0000	.000000	.0000	10.00000	52.17
32	120.00	.0000	-.000579	1.0000	1.00000	48.88
33	120.00	.0000	-.000579	1.0000	1.00000	48.88
34	121.67	.0000	.000000	.0000	10.00000	49.81
35	122.09	.0000	.000000	.0000	10.00000	50.04
36	123.03	.0000	.000000	.0000	10.00000	50.56
37	123.84	.0000	.000000	.0000	10.00000	51.01
38	124.50	.0000	.000000	.0000	10.00000	51.38
39	125.15	.0000	.000000	.0000	10.00000	51.74
40	140.66	.0000	.000000	.0000	10.00000	60.35
41	142.38	.0000	.000000	.0000	10.00000	61.31
42	155.97	.0000	.000000	.0000	10.00000	68.86
43	176.97	.0000	.000000	.0000	10.00000	80.53
44	195.13	.0000	.000000	.0000	10.00000	90.62
45	192.50	.0000	.000000	.0000	10.00000	89.16
46	177.22	.0000	.000000	.0000	10.00000	80.66
47	164.58	.0000	.000000	.0000	10.00000	73.64
48	140.63	.0000	.000000	.0000	15.00000	60.34
49	124.52	.0000	.000000	.0000	15.00000	51.39
50	146.97	2.4000	.000000	.0000	10.00000	63.86
51	149.57	2.4000	.000000	.0000	10.00000	65.30
52	134.95	.0000	.000000	.0000	75.00000	57.18
53	136.48	.0000	.000000	.0000	75.00000	58.03
54	139.71	.0097	.000000	.0000	.01700	59.83
55	164.21	.0000	.000000	.0000	.01700	73.44
56	184.08	.0639	.000000	.0000	.01700	84.48

FLUID CAPACITY RATE ELEMENTS
STREAM NO.= 1 NODE NO.= 70 INLET TEMP.= 120.00 F

SECTION	NODE NO.	TOUT F	FLOW	RHOF	HEAT IN.
1	71	120.08	50.0000	81.5370	.000000
2	72	120.50	25.0000	81.5146	.000000

FLUID CAPACITY RATE ELEMENTS
STREAM NO.= 2 NODE NO.= 71 INLET TEMP.= 120.08

SECTION	NODE NO.	TOUT F	FLOW	RHOF	HEAT IN.
1	73	120.13	25.0000	81.5313	.000000
2	74	124.02	25.0000	81.3560	1.793628

FLUID CAPACITY RATE ELEMENTS
STREAM NO.= 3 NODE NO.= 75 INLET TEMP.= 122.26

SECTION	NODE NO.	TOUT F	FLOW	RHOF	HEAT IN.
1	76	123.41	50.0000	81.2881	1.560986
2	77	124.57	50.0000	81.1850	1.276133
3	78	124.68	50.0000	81.1281	.000000

FLUID CAPACITY RATE ELEMENTS
STREAM NO.= 4 NODE NO.= 70 INLET TEMP.= 120.00

SECTION	NODE NO.	TOUT F	FLOW	RHOF	HEAT IN.
1	80	131.79	50.0000	81.0141	.000000
2	81	134.26	50.0000	80.3725	.000000
3	82	134.81	50.0000	80.2359	.000000
4	83	135.02	50.0000	80.2015	.000000
5	84	135.03	50.0000	80.1919	.000000
6	85	135.77	50.0000	80.1578	.000000

APPENDIX B

METHOD OF BEARING ANALYSIS

The following shows the method of analysis used in sizing the pump bearings. The example is for the modified ATM pump, 580745.

A summary of the analysis follows:

Pump Normal Operating Conditions Assumed:

A. Working fluid - 80% methanol - 20% water (by weight)

B. Pump inlet pressure 11-15 psig

C. Pump pressure rise - 31 psid

D. Pump flow rate - 900 lmb/hr

E. Pump outlet temp - $50^{o}F$

F. Pump rotor speed - 11,000 rev/min

G. Estimated pump axial thrust - 2.5 lbf (towards inlet)

Bearing Design Approach

A. The bearing geometry was estimated using extreme low flow rate portions of NASA TN D-6371 otpimum friction design curves. No attempt was made to iterate to a fully optimum minimum friction design.

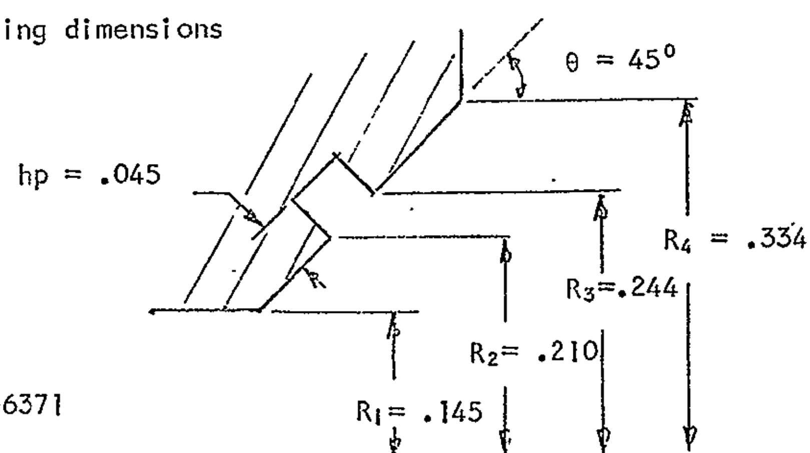
B. Assumptions

1. Design thrust - 5 lbf

2. Bearing fluid temperature - ($50^{o}F$)

3. Running film height - .0005 in. (min)

*4. Bearing dimensions



* Ref. NASA TN D-6371

Definition of Terms

F	Thrust load, LB_f
f_R	Fraction of aera between R_2 and R_3 occupied by pocket, ≈ 1
h_L	Fluid film clearance at lands, in.
h_p	Fluid film clearance at pocket, in.
M_f	Friction torque, in-lb_f
N	Rotor speed, rpm
P	Power loss, watts
p	Fluid supply pressure in pocket, psi
Q	Bearing flow rate, in^3/sec or LBm/hr
R_1	Inner land, inner radius, in.
R_2	Inner land, outer radius, In.
R_3	Outer land, inner radius, in.
R_4	Outer land, outer radius, in.
X_2	R_2/R_1
X_3	R_3/R_1
X_4	R_4/R_1
μ	Fluid dynamic viscosity, LB_m/sec-in or LB_f sec/in^2
ρ	Fluid density, LB_m/in^3
θ	Cone half angle, deg.

Calculations

1. Required pocket fluid pressure

$$p = \frac{2F}{\pi(R_4^2 + R_3^2 - R_2^2 - R_1^2)} \qquad (1)$$

Where $F = 5\ LB_f$

$R_1 = 0.145$ in.

$R_2 = 0.210$ in.

$R_3 = 0.244$ in.

$R_4 = 0.334$ in.

Resulting in

$\underline{p = 30\ psi}$

2. Resulting mass flow rate (loaded bearing)

$$Q = \frac{\pi h L^3 p \sin\theta}{6\mu} \left(\frac{1}{\ln \frac{R_2}{R_1}} + \frac{1}{\ln \frac{R_4}{R_3}} \right) \qquad (2)$$

Where $h_L = 0.0005$ in.

$p = 30$ psi

$\theta = 45$ deg.

$\mu = 2.1 \times 10^{-7}\ LB_f\text{-sec/in}^2$

$R_1 = 0.145$ in.

$R_2 = 0.210$ in.

$R_3 = 0.244$ in.

$R_4 = 0.334$ in.

Resulting in

$\underline{Q = 4.33\ lbm/hr}$

3. Resulting friction torque (loaded bearing)

$$M_f = \frac{\pi\mu\, W_f\, R_1^4}{2\, h_L \sin\theta}\left[X_4^4 - X_3^4 + X_2^4 - 1 + 0.0261\left(\frac{\rho R_1\, W_f h_p}{\mu}\right)^{.75} f_R \frac{h_L}{h_p}\left(X_3^{4.75} - X_2^{4.75}\right)\right] \quad (3$$

Where $\mu = 2.1 \times 10^{-7}$ LB_f - sec/in^2

or 0.81×10^{-4} LBm/in-sec

$W_f = 1.15 \times 10^3$ RAD/Sec

$R_1 = 0.145$ in.

$h_L = 0.0005$ in.

$\theta = 45$ deg

$X_4 = 0.334/0.145$

$X_3 = 0.244/0.145$

$X_2 = 0.210/0.145$

$\rho = 3.1 \times 10^{-2}$ LBm/in^3

hp = 0.045 in.

fR = 1.0

Resulting in

$\underline{M_f = 1.14 \times 10^{-2} \text{ in-LBf}}$

4. Resulting power loss (loaded bearing)

$$P = \frac{M_f \times N}{63024} \times 746 \quad (4)$$

Where $M_f = 1.14 \times 10^{-2}$ in-LB_f

$N = 11{,}000$ rpm

Resulting in

<u>P = 1.485 watts</u>

As a result of the calculation shown in paragraph (1), a layout of the conical bearing fitted into an ATM pump has been completed as shown in Figure 1.

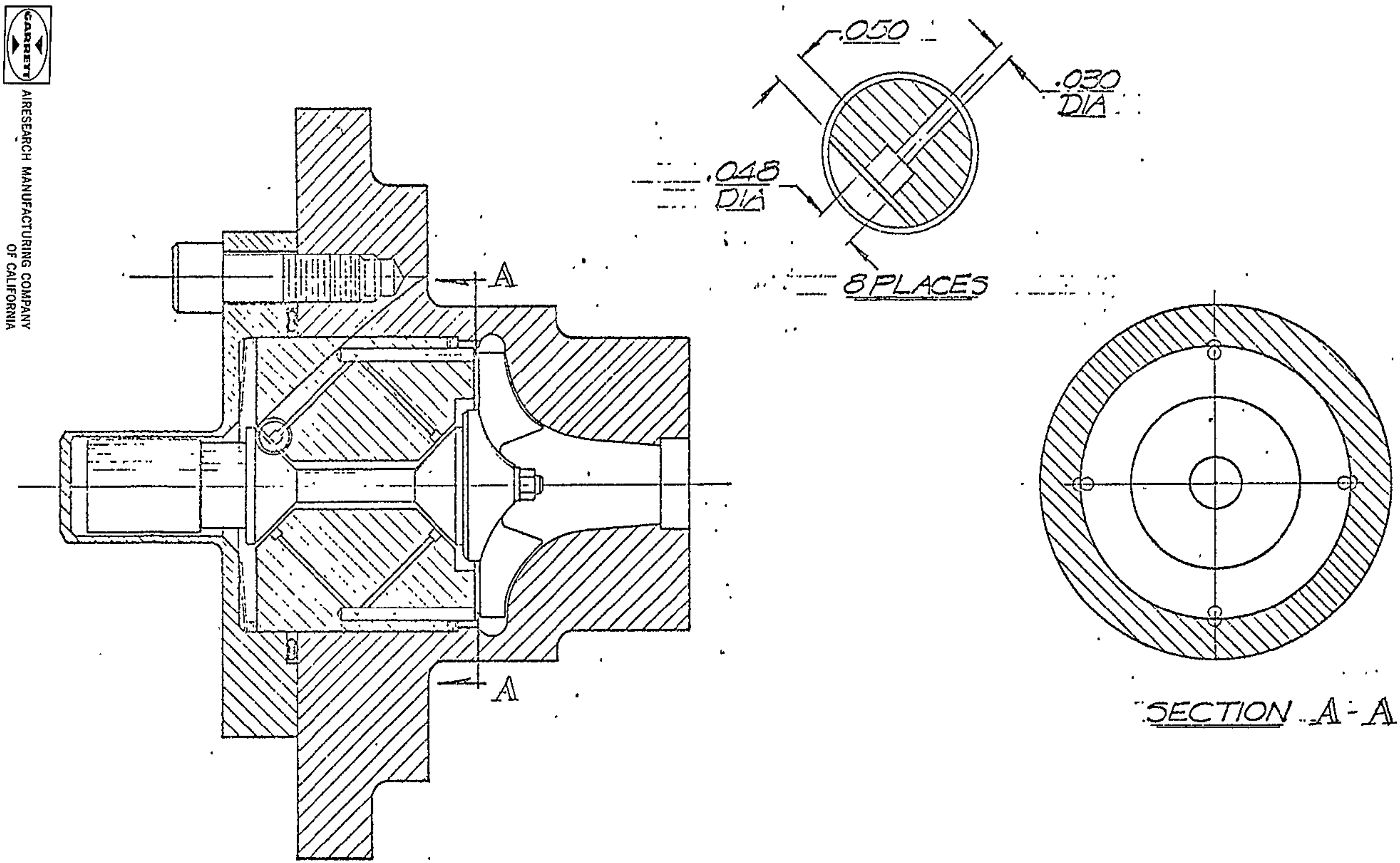


FIG. 1. CONICAL HYDROSTATIC BEARING